Heating·Piping
and Air Conditioning
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Exhaust Steam Utilization Offers

Space-Heating, Air-Conditioning, Drying, Water-Heating, and Process

Economy to Plants and Buildings

By **Perry West***

THE WHY AND HOW of utilizing the heat in exhaust steam for space-heating, water-heating, air-conditioning, drying, water heating and process work will be covered in a practical way in this series. How to make the preliminary study, curves and tables to be used, and examples indicating the results which may be achieved, are included. Coming articles will show how to use waste heat from various other sources.

THE utilization of the heat in exhaust steam offers many opportunities for promoting economy in heating and air conditioning plants and buildings, and in drying and other process work. For instance:

1. The exhaust from ordinary steam-driven machinery contains from 85 per cent to 95 per cent of its available heat, which may in many cases be utilized economically for heating, water heating and in manufacturing processes.

2. Steam may frequently be economically used for air conditioning in summer, and the exhaust may be used for heating and humidifying in winter. When the cost of water is low, exhaust steam-jet refrigeration may be used to advantage.

3. The economy of many drying processes can be improved by use of exhaust instead of live steam.

In studying any problem where it is contemplated to use steam-driven electric-generating or power-producing machinery, utilizing the exhaust steam for other purposes, it is essential to develop curves showing the exhaust steam available and usable for all periods of the operation and from these to determine, after the loads have been as nearly balanced as possible, how much exhaust steam will be wasted and how much make-up steam will be required. After determining the total costs of operation with the use of exhaust steam, these should be compared with the costs of purchasing power and generating steam for use in place of the exhaust. This frequently allows the generation of steam at low instead of high pressure.

It is the purpose of this article to present specific information which will be helpful in planning for the utilization of exhaust steam, including typical problems involving heating, air conditioning, drying, etc.

General Considerations

A pound of saturated steam at 100-lb gage pressure contains 1,189 Btu, consisting of 180 Btu for heating the water from 32 to 212 F (the temperature of steam at atmospheric pressure), plus 970 Btu required to evaporate this water into steam at atmospheric pressure, plus 39 Btu required to raise the pressure and temperature of this steam to the condition of 100-lb pressure.

Conversely, if we start with a pound of steam at 100-lb pressure, we may take 39 Btu out in work, 970 Btu in heat to condense and 180 Btu to cool the condensate to 32 F at atmospheric pressure. The comparative percentages of utilization of heat in such a case are as follows: For work, 3.3 per cent; for condensing, 81.7 per cent; for cooling, 15.0 per cent. Total, 100.0 per cent. If more work is taken out of the steam, a corresponding amount of heat will be thus extracted, so that there will be less left for use in condensing.

As a matter of fact, as soon as the rate of work corresponding to the difference in total heat of the steam at its initial and exhaust pressure is exceeded, a portion of the steam will be condensed, due to this extra work. In other words, a certain amount of work may be "taken out" of the steam without diminishing its quantity at exhaust, so that the same number of pounds of steam fed to an engine or a turbine may be expected to be turned out at the exhaust for re-use in other ways. If more work is done, a percentage of the steam will be condensed and will constitute in the exhaust what is known as the "missing steam" due to work. There is a very narrow limit to the amount of extra work that can be done by the steam, as determined by its expansive characteristics and the efficiency of engines and turbines, so that most of the original heat is left in the steam after its passage through the most efficient machines.

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Assuming a true adiabatic (*i. e.,* no external heat removed or added) expansion of the steam and 100 per cent efficiency of the machine, the greatest amount of heat in the form of external work that can be extracted from a pound of saturated steam between 100 lb gage initial pressure and atmospheric exhaust pressure is 150 Btu. In this case the "missing"steam due to work would be $(150-39) \div 970 = 11.5$ per cent; the heat that could be taken out in condensing the remaining steam would be $970 - 111 = 859$ Btu, and the heat that could be taken out in cooling the condensate to 32 F would remain 180 Btu. The percentages of utilization would be: For work, 12.6 per cent; for condensing, 72.3 per cent, and for cooling, 15.1 per cent.

If we choose to go below atmospheric pressure, more work may be extracted and at 29 in. of vacuum, corresponding to about 0.5 lb per sq in. absolute pressure (about the lowest practical operating pressure for steam-driven prime movers) about 333 Btu per lb, or over twice the above amount, can be extracted for work. The percentages would then be: For work, 28 per cent; for condensing, 68 per cent; for cooling, 4 per cent. Total, 100 per cent.

Similar conditions apply for higher pressure steam, except that the percentage available for work increases slightly. As an example, with steam at 400 lb saturated, the percentages (down to atmospheric pressure) are as follows: For work, 19.5 per cent; for condensing, 65.5 per cent; for cooling, 15.0 per cent. Total, 100.0 per cent. The percentages down to 29-in. vacuum are: For work, 33.8 per cent; for condensing, 62.2 per cent; for cooling, 4.0 per cent. Total, 100.0 per cent.

Heat Balance and Adjustment of Back Pressures

All that has been said so far relates to the utilization of exhaust steam at or below atmospheric pressure. As a rule, steam is exhausted at or slightly above atmospheric pressure, but may be exhausted at higher or lower pressures to suit conditions. However, in studying any specific problem, the fixing of the exhaust pressure should be considered in conjunction with a heat balance and the best economic set-up for the entire situation.

The raising of the back pressure on engines and turbines reduces their power production, but generally raises their efficiency as far as the work produced per unit of heat extracted is concerned. If more exhaust is needed for a proper heat balance, it may be obtained in this way, with a resulting increase in over-all efficiency. Conversely, if more exhaust is normally produced than can be used, it may be reduced by lowering the exhaust pressure, with a corresponding increase in efficiency. In raising exhaust pressures, the temperature of the exhaust is increased, and in lowering them, the corresponding temperatures are decreased. It depends on what use is to be made of the exhaust as to the pressure and temperature that can be maintained.

In low-pressure heating the pressure may be anywhere between 15 lb gage and 15 in. of mercury absolute, but is generally from 1 to 2 lb above atmosphere. In some drying and process work, higher pressures are required and may be obtained either by raising the back pressure or by bleeding the necessary steam out of a turbine or a compound engine somewhere between the throttle and the exhaust outlet, to suit conditions. These questions of heat balance and adjustment of back pressures will be discussed later under the various problems involving specific applications.

These characteristics of steam illustrate very forcibly what happens when it is used in steam turbines, engines, pumps and other steam-driven apparatus for generating power or doing other work and shows what a small percentage of the heat energy in the steam can be used for work. The steam emitted from such units is what is generally termed *exhaust steam* and, as can readily be seen, still contains most of its original heat. It is the utilization of this heat energy with which we are concerned.

It is not to be understood that the heat units used for work are absolutely all that are taken out of the steam in its passage through such engines, turbines, etc., as mentioned, as there are some additional heat losses from friction, condensation, radiation, etc.

These are comparatively small, however, so that the percentages given are substantially correct, at least for all practical purposes.

Some Uses for Exhaust Steam

It can be seen that the greatest field for the utilization of this waste heat is in the condensation of the exhaust steam; some of the most common methods of such utilization are in heating, drying, air conditioning, water heating and in manufacturing processes. The small percentage

Engine room of school using exhaust steam for heating

of heat to be derived from cooling the condensate is rarely found to be useful, especially where the condensate is to be returned to the boiler and would require to be reheated.

Some of the other methods in common use for the utilization of the waste heat from exhaust steam are low-pressure turbines, steam-ejector and absorption refrigeration, water syphons and pulsometer pumps.

One of the first considerations is feed-water heating, a continuous requirement so long as the plant is in operation, but, inasmuch as this requires only about 2 per cent of the available waste heat for plants returning all of the condensate to the boiler and 15 per cent for plants using 100 per cent make-up water, it is a relatively small part of the total heat to be used. Often the next consideration is for the heating of water, as this is also generally a year-round requirement and may in many cases require a considerable part of the available heat.

The use of exhaust steam for heating is a most general and fertile application, but also one which is seasonal only and of such a variable character that it is hard to evaluate and consequently properly to design and prepare for.

The other uses referred to depend for their efficacy of application upon the specific conditions to be met and will be discussed later under particular problems of this character.

Exhaust Steam for Heating

In the use of exhaust steam for heating there are a number of variables and heat balances which must be considered. First, the *loads* on the engines, turbines, pumps and other apparatus from which the exhaust is to be derived are variable; the quantity of exhaust not only varies for this reason but because the *steam rate* of these machines varies according to their loads. Generally speaking, there are daily, seasonal, weather-responsive, and other periodic variations for particular days of the week and for Sundays, holidays, etc.

Different classes of buildings and plants have their peculiar load variations and some classes of machines have greater variations than others. For instance:

Schools may or may not have evening classes, summer sessions, or all-year classes.

Office buildings may or may not have all or part night service, Sunday or holiday service.

Refrigerating machines generally have a comparatively steady load while elevators have perhaps the most fluctuating.

Dark days require heavier lighting loads, extremely cold days cause extra power loads for heating and air conditioning, and extremely hot days may produce similar results.

The heating requirements of buildings vary, depending upon the weather, wind, sun and the uses to which the buildings are put.

Substantially all buildings are heated for a certain portion of the day or day and evening and are allowed either to cool normally or to a certain temperature at night, or for the remaining unoccupied hours. If exhaust steam is available it is of course more economical to keep the temperature constant 24 hours per day, but, in considering any problem where exhaust may otherwise be eliminated during the unoccupied hours, it is more economical to allow the building to cool to at least 55 F than to generate steam for heating alone.

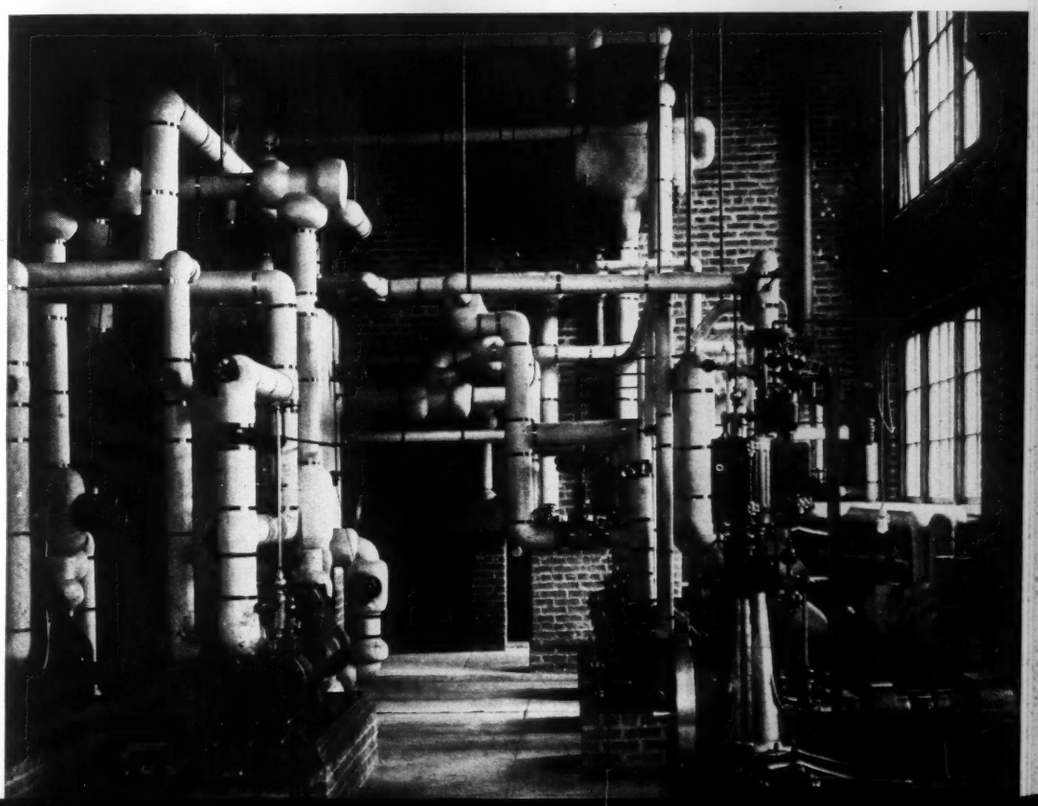
Heating requirements have regular daily variations beginning with a heavy load for heating-up in the morning, usually decreasing as the building becomes heated and the day warms up, and then increasing as the day cools off towards evening. There are also variations due to Sundays, holidays and special occasions, periodic occupancy, etc.

The economy and efficacy of the use of exhaust for heating depends in each specific case on how the data representing the exhaust produced can be fitted into that representing the heating requirements, taking into consideration all of the variables involved.

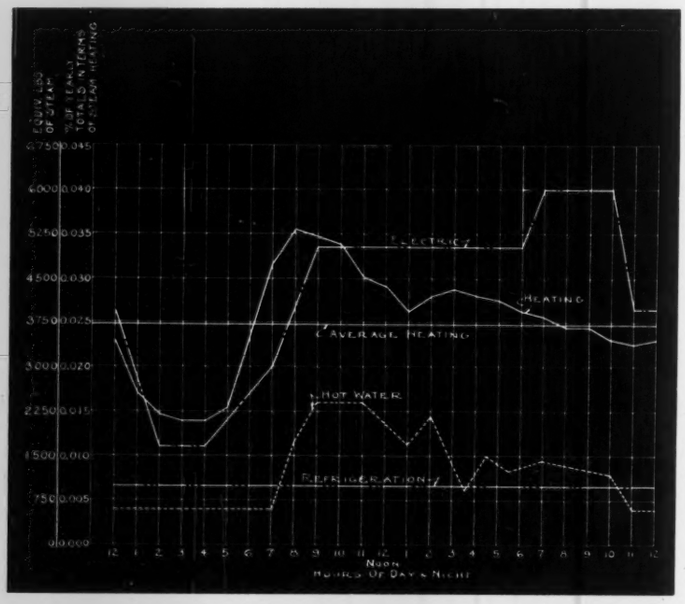
Studying Heating Loads and Exhaust Supply

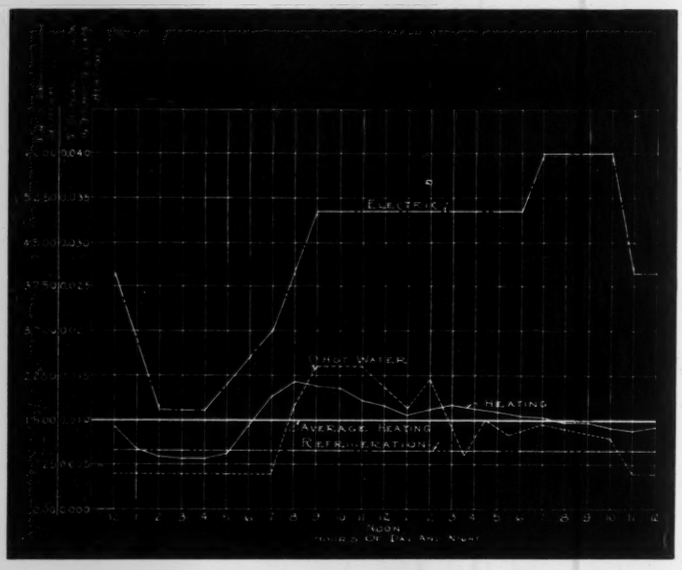
A logical way to study and adjust heating loads and exhaust supply is by means of comparable curves showing the synchronous variations in these quantities of steam required and available. Load conditions and the resulting quantities of exhaust must be computed for the various hours of the day and night, separately for the different seasons and for Sundays, holidays and special days; similar computations must be made for the steam required for heating.

Generally speaking, one set of curves for each of the winter, spring and fall and summer seasons, Sundays and holidays is sufficient, the winter curves covering December to March, the spring and fall curves including



Pump room and auxiliary piping in large school





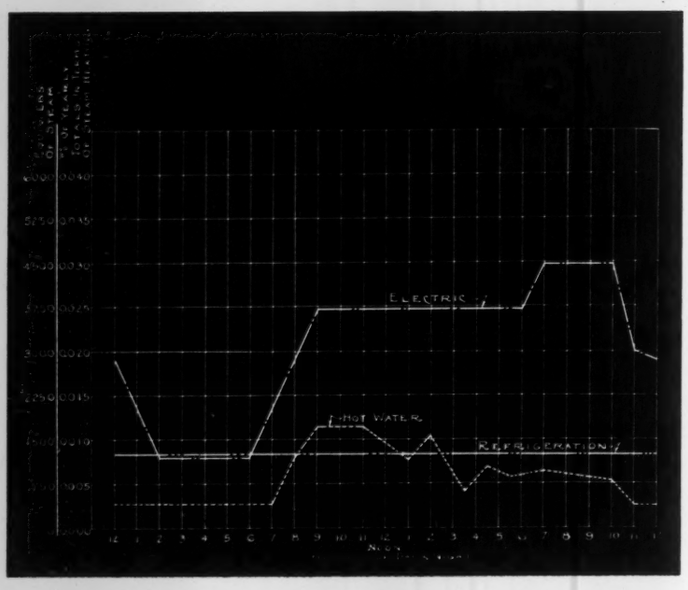


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Fig. 1—Heating, electric, hot-water and refrigerating loads for a typical 250-room hotel for the winter months of December to March

Fig. 2—Heating, electric, hot-water and refrigerating loads for a typical 250-room hotel for the fall and spring months October, November, April and May

Fig. 3—Electric, hot-water and refrigerating loads for a typical 250-room hotel for the summer months June to September

April, May, October and November, and the summer curves, June to September. These curves may be based on lb of steam per hr available and required or upon the basis of the steam per hr available and required in percentage of the total steam required for heating. The former has the advantage of facilitating the balancing of the two factors on a per lb per hr basis, while the latter has the advantage of better comparison with similar problems charted on the same basis.

Figs. 1, 2 and 3 show curves for a typical 250-room hotel on the latter basis; Fig. 4 shows curves for a typical office building of about 9,000,000 cu ft and 80,000 sq ft of steam radiation and Fig. 5 shows curves for a typical manual training high school of about 2,680,000 cu ft and 73,500 equivalent sq ft of steam radiation with elevators and electric generating plant, on the former basis. Fig. 6 shows curves for a typical 50-family apartment building containing about 8,750 sq ft of radiation and two electric elevators.

Method of Developing the Curves

Load curves in kw or hp for each of the 24 hours per day for a typical winter load, a typical spring and fall load, a typical summer load and for typical Sunday, holiday and special day loads, are first developed. Next, these are converted into their corresponding available exhaust steam curves by multiplying each hourly load by the steam rate of the unit, or units, corresponding to the loads carried and deducting from each product the quantity of steam lost in passing through the unit. By the quantity of exhaust thus lost is meant the amount which disappears as steam and passes out as condensate, according to the heat that is extracted from the steam in the form of work, plus the heat lost by radiation and condensation.

These heat losses and the resulting steam loss may be figured as follows: The heat equivalent of a horsepower-hour is 2,546 Btu, while the heat equivalent of a kilowatt-hour is 3,415 Btu. If 2,546 is divided by the steam rate of the unit in lb of steam per hp-hr, or 3,415 is divided by the steam rate per kw-hr, the result will be the number of Btu lost per lb of steam due to work.

In either case the steam rate must be in terms of lb per hr per actual unit of external work done by the steam. In the case of engines this rate is usually given in terms of lb of steam per indicated hp-hr, which is substantially equal to the work done by the steam. If it is given in terms of lb per kw-hr delivered by the generator it may be converted to lb per kw-hr of work done by the steam by multiplying the rates thus given by the combined efficiency of the engine and generator.

After determining the heat equivalent of the external work done by the steam per lb, the next step is to de-

duct this from the total heat per lb of steam fed to the unit and from this remainder to deduct the heat per lb of the liquid (as shown by steam tables) for steam at the pressure of the exhaust. From this last remainder deduct 2 per cent of the total heat in the steam for radiation and condensation losses.

This gives the heat per lb left in the steam above its temperature of condensation at the exhaust pressure and by comparison with the latent heat of steam at this pressure it may be determined whether steam has actually disappeared as such or, in isolated cases, it may show that the actual steam has not only suffered no diminution but may actually be all present or even superheated.

If the heat remaining is less than the latent heat, some steam has dissapeared as such; if it is equal, the steam is all there and saturated; and if it is greater, the steam is all there but superheated.

If steam has disappeared, the quantity missing may be found by deducting the heat remaining in the exhaust per lb from the heat of vaporization and dividing this remainder by the heat of vaporization to give the percentage of missing steam. By multiplying this percentage by the total steam used and deducting the result from the total steam, the available steam can be found.

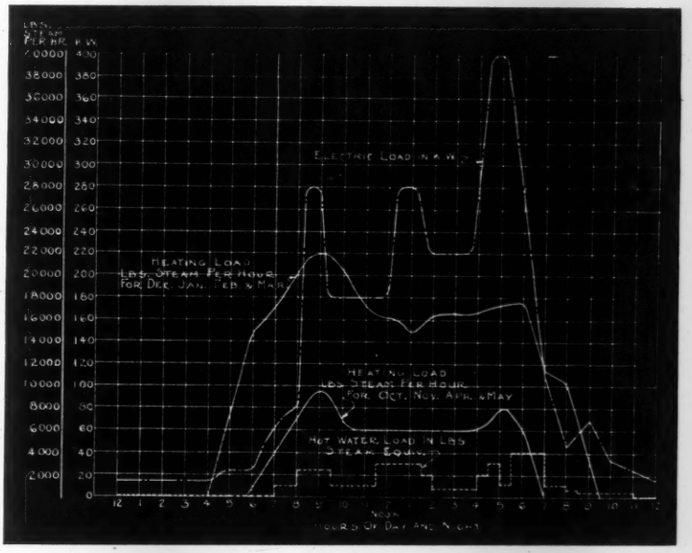
Example: A turbo-generator operating on a load of 200 kw uses 40.0 of steam per kw-hr delivered by the generator when using saturated steam at 150 lb gage and exhausting against atmospheric pressure. The over-all efficiency of the turbine and generator is 0.873. Determine the quantity of exhaust per hour available for heating.

Solution: The steam rate of the unit per kw-hr of external work actually done by the steam is $40.0 \times 0.873 = 34.92$ lb of steam per kw-hr.

The heat equivalent of this work per lb of steam fed to the turbine is $3415 \div 34.92 = 97.79$ Btu.

The total heat of a lb of saturated steam at 150 lb gage is 1195 Btu, and deducting 97.79 from this leaves 1097.21 Btu.

The heat of the liquid per lb of steam at atmospheric pressure is 180 Btu and deducting this from 1097.21 leaves 917.21. Deducting from this 2 per cent of 1195 = 23.9 for heat lost by radiation and condensation leaves 893.31 Btu per lb of steam originally fed to the turbine.


ELECTRIC LOAD IN K.W.
HEATING LOAD
LBS. STEAM PER HOUR
FOR DEC. JAN. FEB. & MAR.
HEATING LOAD
LBS. STEAM PER HOUR
FOR OCT. NOV. APR. & MAY
HOT WATER LOAD IN LBS
STEAM EQUIV.
HOURS OF DAY AND NIGHT

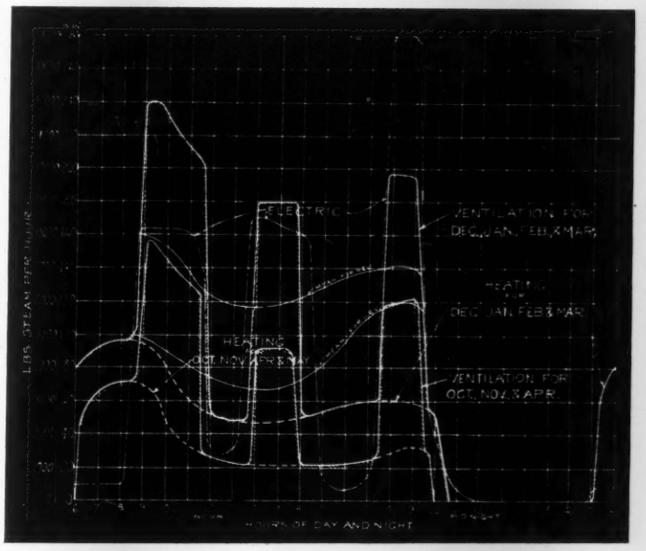


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Fig. 4—Heating, electric and hot-water loads for a typical office building of about 9,000,000 cu ft and 80,000 sq ft of steam radiation for the winter months December to March and the fall and spring months of October, November, April and May

Fig. 5—Heating, ventilating and electric loads for a typical manual training high school of about 2,680,000 cu ft, 20,000 sq ft of direct radiation and an equivalent of 53,500 sq ft in air heaters, for the winter months December to March and the fall and spring months of October, November, April and May

Note: The heating load for schools generally may be estimated as follows:
150 to 200 sq ft of direct radiation per class room.
400 to 500 sq ft equivalent for indirect radiation in air heaters for ventilating per class room.
300 to 400 lb of steam per season per sq ft of direct radiation.
75 to 100 lb of steam per season per equivalent sq ft of indirect radiation. During December, January, February and March the direct-heating load is on for 105 days, and the fan-heater load is on for 80 days. During October, November and April the direct-heating load is on for 85 days and the fan-heater load is on for 60 days. All on the basis of the latitude of New York City with 5300 degree days per year and corrected for other localities in proportion to the degree days.

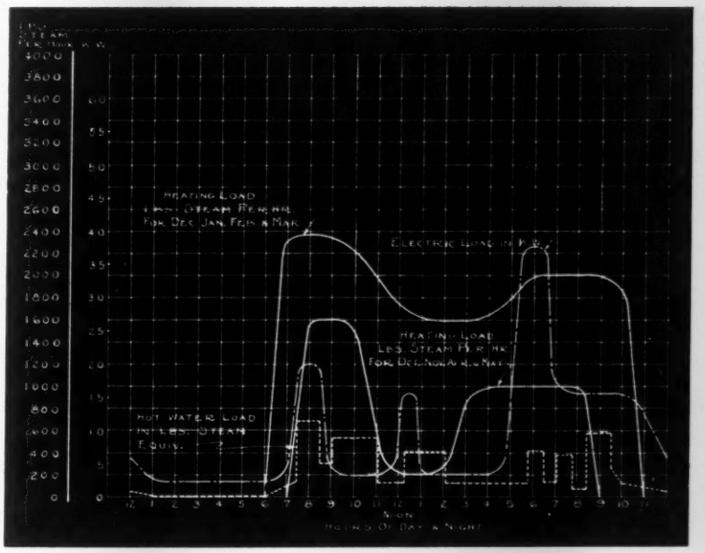


Fig. 6—Heating, electric and hot-water loads for a typical 50-family apartment building, containing about 8750 sq ft of radiation and two electric elevators

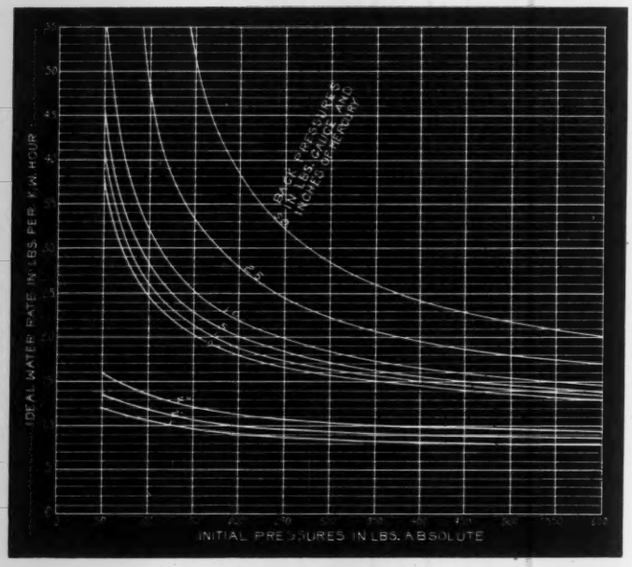


Fig. 7—Ideal steam consumption or water rates for steam turbines and engines in pounds per kilowatt hours for different initial absolute pressures and with different back pressures measured above atmosphere in gage pressure and below atmosphere in inches of mercury absolute

The heat of vaporization for steam at atmospheric pressure is 970.4, which being less than the remaining heat of 893.31 shows that the steam equivalent of 970.4 — 893.31 = 77 Btu has been lost. The percentage of missing steam is therefore 77 ÷ 970.4 = 7.9. The total quantity of steam fed to the turbine per hour was 200 × 40 = 8000 lb. The missing steam in the exhaust is 8000 × 7.9 per cent = 632 lb and the remaining exhaust available for heating is 8000 — 632 = 7368 lb per hr.

With this information as to the available exhaust for each hour of the day and night for a typical day of each season, Sundays, holidays and other special days, curves can be made up corresponding to the data.

If the exact steam rates of the units are not available, very close estimates may be made from Fig. 7, which shows curves giving the ideal steam or water rate for units of various sizes when operating under various initial steam and exhaust conditions; Table 1, which shows the mechanical efficiencies of engines and turbines, the over-all efficiency of generators and the factors by which to multiply the ideal water rate to determine actual water rates for engines and turbines of various sizes; and Table 2, which gives the factors by which to multiply the actual water rates at full load to determine the estimated water rates at fractional loads. Table 3 gives data from which electric loads may be estimated in conjunction with the curves displayed for different types of buildings. From these data performance curves of the various units may be plotted.

Table 1—Approximate Full-Load Efficiencies for Non-Condensing Engine and Multi-Stage Turbo-Generator (Geared Type)

Rated Capacity at Full Load, Kw	Mechanical Efficiencies of Engine Units at Full Load	Mechanical Efficiencies of Turbine Units at Full Load	Overall Generator Efficiencies at Full Load	Factors for Converting Ideal Water Rates to Actual Water Rates (For Turbines only)*
100	0.900	0.930	0.932	2.22
200	0.905	0.935	0.934	2.00
300	0.910	0.940	0.936	1.85
400	0.915	0.942	0.938	1.74
500	0.920	0.945	0.940	1.64
600	0.925	0.947	0.942	1.56
750	0.930	0.950	0.944	1.54
1000	0.935	0.952	0.945	1.49
1250	0.940	0.955	0.947	1.49
1500	0.945	0.957	0.950	1.49
2000	0.950	0.960	0.954	1.45
		Condensing Turbines		
500		0.950	0.940	1.45
600		0.952	0.942	1.44
750		0.954	0.944	1.43
1000		0.956	0.945	1.43
1250		0.958	0.947	1.43
1500		0.960	0.950	1.43
2000		0.960	0.954	1.43
				(For Non-Condensing Engines only)
Single-valve automatic engine units up to 400 kw				1.64
Four-valve automatic engine units up to 400 kw				1.42
Unaflow engine units up to 400 kw				1.20

*These factors are for indicated engine or brake turbine loads and do not take into account the mechanical efficiency of engine units or generator efficiencies for either engine or turbine units; but do include the mechanical efficiency of turbines.

Table 2—Factors by Which to Multiply Actual Water Rates at Full Load to Obtain Estimated Water Rates at Fractional Loads on Non-Condensing Engine and Turbine-Driven Electric Generating Units*

Size and Kind of Unit	Load ¼	½	¾	Full
Single-valve engines up to 400 kw	1.32	1.09	1.03	1.00
Four-valve engines up to 400 kw	1.30	1.08	1.03	1.00
Unaflow engines up to 400 kw	1.10	1.02	0.980	1.00
Geared turbines up to 600 kw	1.57	1.25	1.05	1.00
Geared turbines 600 to 2,000 kw	1.50	1.20	1.04	1.00

*These values are for indicated or brake loads and do not take into account variations in generator efficiencies under various load conditions. Generator efficiencies vary from 85 per cent at one-quarter load to 95 per cent at full load.

Example: A 200-kw turbo-generator is operating on saturated steam at 150 lb gage and exhausting against atmospheric pressure. Determine its steam rate in lb per kw-hr delivered at the generator at ½ load.

Solution: From. Fig. 7 the ideal rate is found to be 20.15 lb. From Table 1 the factor by which to multiply the ideal rate to determine the actual rate at full load is found to be 2.00. Multiplying this gives 20.15 × 2.00 = 40.30 lb of steam per kw-hr delivered by the turbine.

From Table 1 the overall efficiency of the generator is found to be 0.934. Dividing 40.30 by 0.934 gives 43.15 lb per kw-hr delivered by the generator at full load.

From Table 2 the factor by which to multiply this full-load water rate to determine the water rate at ½ load is found to be 1.25. Multiplying this out gives 43.15 × 1.25 = 53.94 lb per kw-hr delivered by the generator at ½ load, which is the answer sought.

Similar data can be worked out for engines in the same manner, but it should be noted that the factors given in Table 1 by which to multiply ideal water rates for engines to determine actual water rates do not include the mechanical efficiency of the engine. The actual water rates thus obtained must be divided by the combined efficiencies of the engine and generator in order to determine the actual water rate per unit of power delivered by the generator. This combined efficiency is obtained by multiplying the mechanical efficiency of the engine by the over-all efficiency of the generator. For example, if in the last problem the unit had been an engine instead of a turbine, the combined efficiencies would have been 0.905 × 0.934 = 0.845 and this figure would have been used instead of 0.934.

Table 3—Connected Loads in Watts Per Square Foot of Floor

	Light	Power
Office Buildings	0.80	0.75
Hotels	0.80	0.60
Hospitals	1.00	0.50
Residences	1.50	0.25

Kilowatt Hours Used Per Year Per Square Foot of Floor

	Light	Power
Office Buildings	1.50 to 1.75	0.50 to 1.00
Hotels	1.25 to 1.50	2.50 to 3.50
Hospitals	1.00 to 1.25	1.00 to 1.50
Residences	0.25 to 0.50	0.10 to 0.25

Computing the Thickness of Pipe

Article No. 2 on

The New Code *for* Pressure Piping

By **Sabin Crocker***

THIS IS a continuation of the "preview" of the Proposed American Standard Code for Pressure Piping which started in the January issue. The present installment deals with the formula for pipe-wall thickness, explaining the basis of the formula and differentiating between the requirements of individual sections of the Code. The formula serves to reconcile in an ingenious manner the empirical rules of the old order with a rational system of computation, thus harmonizing past practice with a new system which can be extended to fields beyond the scope of empirical rules.

ONE of the important tasks undertaken by the sectional committee on Code for Pressure Piping was setting up a formula for use in computing safe thicknesses of pipe for any application in any of the several Code sections. This problem was referred to a special sub-group appointed for the purpose of analyzing formula requirements from all angles. It was felt that in a safety code, allowable stress factors (S values) should be set with reference to actual thicknesses measured on inspection rather than on average or "nominal" thicknesses given in published pipe schedules. A modified version of the Barlow formula seemed the best way to obtain reasonable simplicity, and at the same time provide for the variety of materials and end connections in common use. Precedent for the adoption of such a formula was found in the *A. S. M. E.* Power Boiler Code and in the proposed pipe standard of *A. S. A.* Sectional Committee *B36*.[1] Further, values of wall thickness computed by use of the Barlow formula have been shown by considerable published data to be as close to theoretically correct as is warranted in the use of commercial pipe.

In order to fit the formula to the considerable variety of materials available for use in pipe, it became necessary to assign appropriate S values to each of these materials on a basis of load-carrying ability. Also, to provide an allowance for mechanical strength and/or corrosion, and at the same time accommodate the multiplicity of end connections now in common use, provision was made in the formula for a constant C representing an arbitrary increase in wall thickness, the amount of increase depending on whether the pipe was threaded, van stoned, grooved, welded or otherwise attached. While consideration of the type of end connection on the wall thickness of pipe has not been accorded in other Codes, the increasing use of welded and other forms of plain end attachment plainly warranted the forward step. Certain definite restrictions in applying the formula seemed required to govern the maximum allowable temperatures for brass, copper, and cast iron; the minimum thickness schedules where pipe is threaded; and the lowest usable values of service pressure P. The intent of the two latter restrictions is to insure having sufficient mechanical strength in low- and medium-pressure pipe, and to provide a margin for water hammer in the case of cast-iron pipe.

The formula as finally worked out by the sub-group and adapted to the requirements of the Section on Power Piping Systems is as follows:

"Thickness of Pipe. (Power Piping Systems) (a) In determining the minimum thickness on inspection to be used for pipes at different pressures and for temperatures not exceeding 750 F for steel or wrought-iron pipe, 406 F for brass and copper pipe, and 450 F for cast iron, *Formula 1* shall be used.

$$t_m = \frac{PD}{2S} + C \qquad (1)$$

where t_m = minimum pipe wall thickness in in., allowable on inspection.

P = maximum internal service pressure in lb per sq in. gage (plus water hammer allowance in case of cast iron pipe conveying liquids, see *Par.* 122 (*d*).

D = actual outside diameter of pipe, in in.

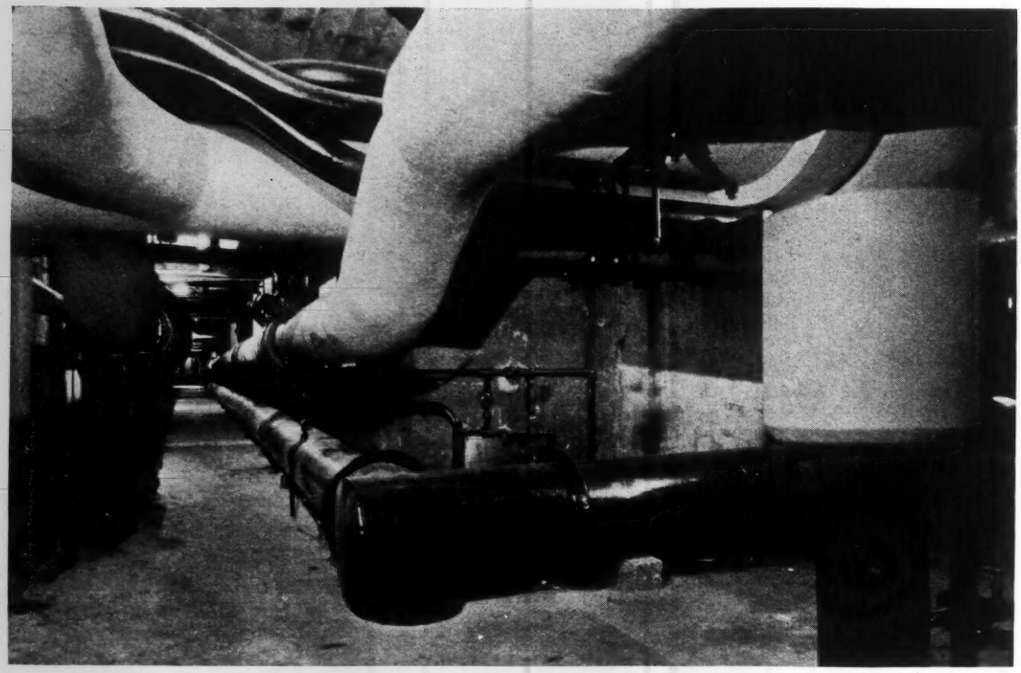
S = allowable stress in material, in lb per sq in.

C = factor for allowance for threading, mechanical strength and/or corrosion.

"(b) The value of C used in *Formula 1* shall not be less than that given for the respective material in Table 1.

*Engineer, The Detroit Edison Company, and Member of Board of Consulting and Contributing Editors. Among other standardization committee affiliations Mr. Crocker is Chairman of Subcommittee 1 on Plan, Scope and Editing of *A.S.A.* Sectional Committee *B31* on Code for Pressure Piping. This article continues the series on piping standardization which started in the September, 1933, issue.

[1]See pp. 502 to 504, HEATING, PIPING AND AIR CONDITIONING, October, 1933.



Intersection of main longitudinal trench with lateral trench through which all lines enter the underground system, at Rockford Drop Forge Co., Rockford, Ill. Pipe bends and welding turns used wherever possible on lines 1½ in. and larger. All work welded except where connected to equipment. The large pipe in the lower central portion is welded sheet iron pipe, No. 10 gage

Photos courtesy Lockwood Greene Engineers, Inc.

Table 1—C Values

Material	Value of C[1] in Inches
Cast-iron pipe, centrifugally cast or cast horizontally in green sand molds	0.14
Cast-iron pipe, pit cast	0.18
Threaded-steel, wrought-iron, or non-ferrous pipe	Depth of thread in in.
Grooved-steel, wrought-iron or non-ferrous pipe	Depth of groove in in.
Plain end[2] steel, wrought-iron or non-ferrous pipe, for 1-in. size and smaller	0.05
Plain end[2] steel, wrought-iron or non-ferrous pipe, for sizes above 1 in.	0.065

[1]The values of C stipulated above are such that the actual stress in the wall of the pipe is less than the values of S set out below as applicable in *Formula* 1.

[2]Plain end includes pipe joined by flared compression couplings, lap or van stone joints, and by welding, *viz.* by any method which does not reduce the wall thickness of the pipe at the joint."

"(c) The value of S in *Formula* 1 shall not exceed that given in Table 2 for the respective material:

Table 2—S Values

Material	Value of S in Lb per Sq In.
Steel pipe, seamless, *Grade C* (*A.S.T.M. A106*)	14,000
Steel pipe, seamless, *Grade B* (*A.S.T.M. A106* and *A53*)	11,500
Steel pipe, seamless, *Grade A* (*A.S.T.M. A106* and *A53*)	9,000
Steel pipe, *Class 1* fusion-welded (*A.S.T.M.* specification under preparation)	**
Steel pipe, fusion-welded, *Grade B* (*A.S.T.M. A139*)	**
Steel pipe, fusion-welded, *Grade A* (*A.S.T.M. A139*)	**
Steel pipe, ordinary fusion-welded (*A.S.T.M. A134*)	**
Steel pipe, resistance-welded (*A.S.T.M. A135*)	**
Steel pipe, lap-welded (*A.S.T.M. A53*)	7,000
Steel pipe, butt-welded (*A.S.T.M. A53*)	**
Steel or wrought-iron pipe, riveted joint (*A.S.T.M. A138*)	**
Wrought-iron pipe, lap welded (*A.S.T.M. A72*)	5,300
Wrought-iron pipe, butt-welded (*A.S.T.M. A72*)	4,500
Brass pipe, seamless (*A.S.T.M. B43*)	4,500
Copper pipe, seamless (*A.S.T.M. B42*)	4,000
Copper tubing, seamless (*A.S.T.M. B75*)	4,000
Copper tubing, seamless (*A.S.T.M. B88*)	4,000
Cast-iron pipe centrifugally cast or horizontally cast in green sand molds (*F.S.B. WW-P421*)	6,000
Cast-iron pipe, pit-cast (*A. W. W. A.*)	4,000

**The setting of allowable stresses for these types of pipe is under consideration by a special subgroup on S Values, the problem hinging on agreement as to reasonable joint efficiencies for the various types of welds. In general, the basic principle followed is to set S values equal to the ultimate strength of the material times the joint efficiency, divided by a factor which, in the case of power piping, is approximately 5, *viz*

$$S = \frac{\text{(ultimate strength of material)} \times \text{(joint efficiency)}}{\text{approximately 5}}$$

"(d) In the case of cast-iron pipe the minimum values of the water-hammer allowance to be added to P are given in Table 3.

Table 3—Water-Hammer Allowance for Cast-Iron Pipe

Pipe Diam. in Inches (Sizes, Inclusive)	Water-Hammer Allowance
4 to 10	120
12 to 18	110
20	100
24 to 30	95
36 to 48	90
54 to 84	85

"(e) In *Formula* 1 the value of P shall not be taken at less than 100 lb per sq in. for any conditions or material.

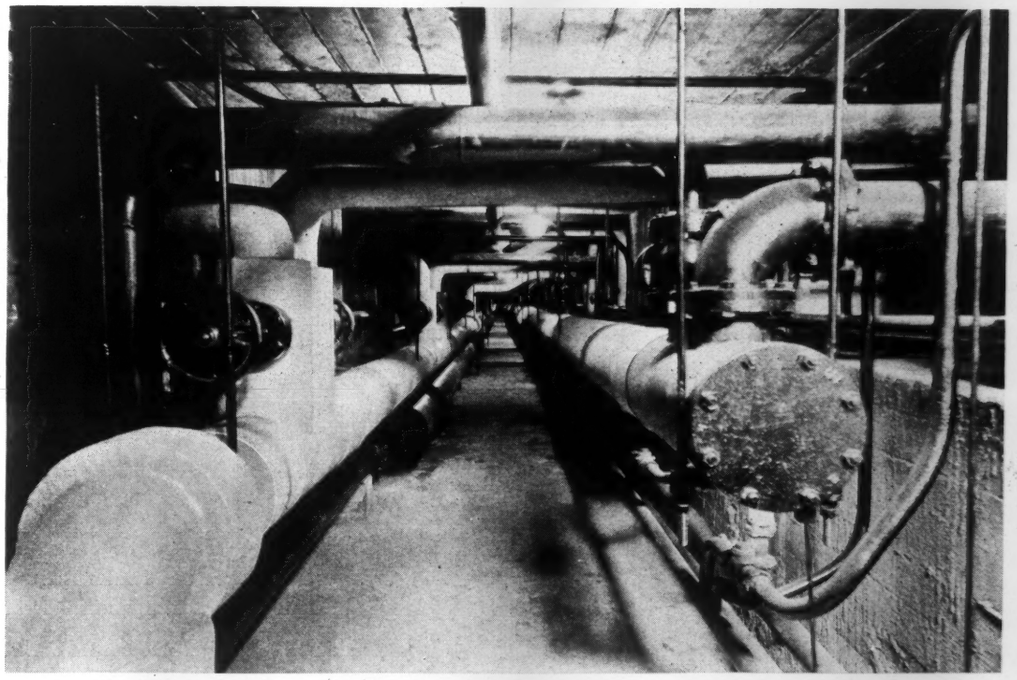
"(f) Steel or wrought-iron pipe lighter than standard weight shall not be threaded.

"(g) While the thickness given by *Formula* 1 is theoretically ample to take care of both bursting pressure and material removed in threading, it is mandatory that where pipe is threaded and is used for steam pressure of 250 lb per sq in. and over, or for piping under pressure in excess of 125 lb per sq in. with water temperature of 220 F and over, a weight and quality at least equal to the 600-lb class of *A. S. T. M.* Specification *A106* be used to furnish added mechanical strength."

Since each of the several sections of the Code covers a different class of piping for conveying diverse fluids with various degrees of hazard, it is manifest that C and S values for use in the formula may properly vary between Code sections for the same material, depending on the fluid conveyed and the service conditions involved. For instance, in oil refinery practice where lines are frequently disassembled for cleaning and inspection, a higher value of allowable stress for a pipe material is permissible than for the same material used in steam-power-plant piping where once the line is erected, it may not be disassembled again until the plant has outlived its usefulness.

Diversity of S Values

In Table 4 are compiled the S-values for different pipe materials from all sections of the Code, together with a listing of the corresponding tensile strengths, joint efficiencies, and other factors with respect to which the S-values were set.



Underground tunnel below floor of drop forge plant, furnishing following services to 41 steam and board hammers through system of lateral trenches: 10-in. high-pressure steam, 12-in. low-pressure steam, 30-in. 1½-lb air, 5-in. 30-lb air, 4-in. 100-lb air, 2½-in. fuel oil, high-pressure and low-pressure condensate return, refrigerated drinking water, drinking water drains, and service water

Due to differences in application of the product contemplated under the various Code sections, as well as to variations involving the kind of material and its tensile strength, method of manufacture, etc., absolute consistency throughout the table cannot be attained. It should be borne in mind, therefore, that the assignment of values for joint efficiencies and other factors was merely a guide for use in setting up for diverse service conditions reasonable *S*-values, which in the final selection were predicated to a considerable degree on actual experience with the product in service.

Then also the degree to which secondary stresses and/or hazard is involved has a bearing on setting up a systematic schedule of *S*-values. For instance, in the

Table 4—Basis of Allowable Stress (S Values) Listed for Use in Pipe Wall Thickness Formula*

Code for Pressure Piping
November 1933 Draft

			Power			Gas and Air			Oil			District Heating		
	Specification Designation	Tensile Strength psi (TS)	Factor of Safety **	Joint Effic % (*E*)	*S* Value psi	Factor of Safety **	Joint Effic % (*E*)	*S* Value psi	Factor of Safety **	Joint Effic % (*E*)	*S* Value psi	Factor of Safety **	Joint Effic % (*E*)	*S* Value psi
Steel Pipe, Seamless, Grade A	A106, 5L	48,000	5.3	100	9,000	3.5	100	13,700	2.8	100	17,400	5.3	100	9,000
Grade B	A106, A53	62,000	5.4	100	11,500	3.5	100	17,700	.....	.....	.....	5.4	100	11,500
Grade C	A106, 5L	75,000	5.4	100	14,000	3.5	100	21,500	2.8	100	27,300	5.4	100	14,000
Steel Pipe, Seamless, Low Carbon***	A53	48,000	5.3	100	9,000	3.5	100	13,700	.....	.....	.....	5.3	100	9,000
Steel Pipe, Seamless, Grade B	5L	60,000	.....	.....	.....	3.2	100	18,500	2.8	100	21,800	.....	.....	.....
Steel Pipe, Class I Fusion Welded														
Grade A	A —	45,000	5.5	90	7,400	.....	.....	.....	.....	.....	.....	.....	.....	.....
Grade B	A —	50,000	5.5	90	8,200	.....	.....	.....	.....	.....	.....	.....	.....	.....
Grade C	A —	55,000	5.5	90	9,000	.....	.....	.....	.....	.....	.....	.....	.....	.....
Steel Pipe, Fusion Welded, Grade A	A139	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
Grade B	A139	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
Grade A	A134	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
Grade B	A134	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
Steel Pipe, Resistance Welded, Grade A	A135	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
Grade B	A135	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
Steel Pipe, Lap-welded	A53, A106	45,000 OH	5.3	83	7,000	3.5	80	10,300	.....	.....	.....	5.3	83	7,000
Steel Pipe, Lap-welded	5L	45,000 OH	.....	.....	.....	3.5	80	10,300	2.8	80	13,100	.....	.....	.....
Steel Pipe, Forge Welded, Grade A	A136	45,000	.....	.....	.....	.....	.....	.....	.....	.....	.....	5.3	83	7,000
Grade B	A136	50,000	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
Steel Pipe, Butt Welded	A53	50,000-B	5.3	53	5,000	3.5	50	7,100	.....	.....	.....	5.3	53	5,000
Steel Pipe, Butt Welded	5L	50,000-B	.....	.....	.....	.....	.....	.....	2.8	51	9,100	.....	.....	.....
Steel or Wrought-Iron Pipe, Riveted Joint	A138	.....	.....	.....	$\frac{TSxE}{5}$	.....	.....	$\frac{TSxE}{4}$	.....	.....	.....	.....	.....	$\frac{TSxE}{5}$
Steel Pipe, Lock Bar	A137	40-50,000	.....	.....	.....	.....	.....	15,000xE	.....	.....	.....	.....	.....	.....
Wrought-Iron Pipe, Lap-welded	A72	40,000	5.3	70	5,300	3.5	79	9,000	.....	.....	.....	5.3	70	5,300
Wrought-Iron Pipe, Lap-welded	5L	42,000	.....	.....	.....	3.5	75	9,000	2.8	81	12,200	.....	.....	.....
Wrought-Iron Pipe, Butt Welded	A72	40,000	5.3	60	4,500	3.5	53	6,000	.....	.....	.....	5.3	60	4,500
Wrought-Iron Pipe, Butt Welded	5L	42,000	.....	.....	.....	3.5	50	6,000	2.8	51	7,600	.....	.....	.....
Brass Pipe	B43	.....	.....	100	4,500	.....	100	6,000	.....	100	7,600	.....	100	4,500
Copper Pipe or Tubing	B42, B75, B88	30,000	7.5	100	4,000	5.5	100	5,500	4.3	100	7,000	7.5	100	5,000
Cast Iron Pipe, Centrifugal Cast	WW-P421	30,000	5.0	100	6,000	5.0	100	6,000	.....	.....	.....	5.0	100	6,000
Cast Iron Pipe, Pit Cast	AWWA	20,000	5.0	100	4,000	5.0	100	4,000	.....	.....	.....	5.0	100	4,000

*Note: Where blank spaces appear in the table, either no values have been assigned or else generally acceptable figures have not as yet been agreed upon. *S*-values only will appear in the Code. The values listed for "Factor of Safety" and "Joint Efficiency" are incidental to setting *S*-values, and subject to discussion.

**Note: Factor of safety with reference to specified tensile strength of material. This is not a true factor of safety, but rather a divisor applied to the tensile strength values. The actual factor of safety is somewhat higher due to the inclusion of *C* values in the formula.

***Note: Yield point 26,500 psi instead of 30,000 for corresponding grade *A106* and *5L*.

case of power piping the secondary stresses induced by thermal expansion may be as high as the bursting stress due to internal pressure, although not necessarily directly additive to the latter, while other rather high bending stresses may be set up by the dead weight between supports. With cross-country gas or oil lines, however, such secondary stresses are of relatively minor importance, and the hazard involved is negligible.

The foregoing considerations are reflected in the *S* value for *Grade A* seamless pipe, which in *Section 3* on Oil Piping Systems is set at 17,400 psi, but in *Section 1* on Power Piping is assigned a value of only 9,000 psi.[2] The oil section contains a further provision that for piping outside refinery limits the *S*-values may be increased 40 per cent beyond those shown in the oil column of Table 4. In the Gas and Air Section provision is made, in the case of cross-country lines outside of corporate limits, for a 25 per cent increase in the *S*-value of cast-iron pipe, and for the use of other materials at service pressures up to 80 per cent of the hydrostatic test pressure applied at the mill.

Diversity of C Values

While the diversity of *C*-values is not as great as exhibited in the case of *S*-values, there are some differences for plain-end pipe betwen Code sections as shown in Table 5.

Aside from the differences noted in Table 5, the *C*-values are essentially the same between Code sections. The values of *C* are stipulated in the formula so that actual bursting stresses will be somewhat lower than the assigned *S*-values, thus providing a margin for secondary stresses, mechanical strength, and/or corrosion. Extended service experience has demonstrated that the practice embodied in the pipe wall thickness formulas given in the various sections of the Code is safe and suitable for the respective operating conditions. Insofar as power piping is concerned, the Boiler Code formulas for pipe-wall thickness are practically identical with that given above for threaded pipe, the Boiler Code having not as yet recognized a distinction between threaded and plain-end pipe.

Maximum Service Temperature for Various Materials

The setting of maximum service temperatures for different metals such as cast iron, wrought iron, steel, copper, and brass has been left to the individual Code sections, and naturally some differences exist due to variations in the nature of the fluids conveyed, the expected life of the equipment, and the opportunities for inspection and replacement of corroded parts. Operating temperatures in oil refineries have reached 1000 to 1100 F for some years back, while the former top limit of 750 F in the power industry is fast moving itself up to 850 F. Such advances in the art will, from time to time, require adjustments in the Code set-up which can easily be made.

Explanation of Paragraphs d, e, f and g

In order to fit a single pipe-wall thickness formula to the variety of materials and methods of end attachment, and to provide sufficient mechanical strength in designing pipe for low service pressures, it was necessary to prescribe the arbitrary allowances set forth in *Paragraphs d, e, f, and g* of the Code quoted above. Through the provisions of these paragraphs, it was possible to adapt the formula to give results consistent with what was known to be safe practice established through long usage.

[2] "Psi" is "pounds per square inch."

Table 5—C-Values for Plain-End Pipe, Inches

	1-in. size and smaller	Sizes over 1-in.
Section 1, Power Piping	0.05	0.065
Section 2, Gas and Air Piping	0.02	0.05
Section 3, Oil Piping	0.05	0.05
Section 4, District Heating Piping	0.05	0.065

(d) Water-Hammer Allowance. The water-hammer allowances for cast-iron pipe listed in Table 3 and similar tables in other sections of the Code are intended to provide a margin between the service pressure and the design pressure (*P*-value) used in the formula sufficient to compensate for hydraulic shock due to surges in flow. For average pipe weights this increase in the *P*-value to compensate for water hammer corresponds in the case of water to an instantaneous change in velocity of about 2 ft per sec. These water-hammer allowances were worked out to bring the formula into line with long established practice in the use of cast-iron pipe as reflected in the working pressure ratings used in specifications such as that of the *American Water Works Association* and *A. S. T. M. A-44*. In general the allowances for water hammer in cast-iron pipe are uniform throughout the Code, except that they are not applicable to vertically-cast pipe in the section on oil piping, where the use of this product is governed by the ratings given in *A. S. T. M. A-44*. The values shown in Table 3 are in reasonably close agreement with the water-hammer allowances used by several large municipalities in the design of their cast-iron water-supply systems; also with the *Federal Specifications Board* requirements for cast-iron pipe.

(e) Minimum Value of P. The purpose in setting a minimum *P*-value of 100 psi for any conditions or material was to insure having sufficient wall thickness to resist mechanical injury and to provide an additional margin for corrosion in thin-walled ferrous pipe. An allowance of this kind must of necessity be somewhat arbitrary.

(f) and *(g) Minimum Wall Thickness for Threading.* Due to the depth of taper thread required for American Standard (Briggs) pipe thread, it is impracticable to thread pipe lighter than "standard weight" irrespective of how low the service pressure may be. In the case of steam pressures 250 psi and over, or water pressures in excess of 125 psi at temperatures above 220 F, the hazard involved through snapping off a pipe in the thread makes advisable the use of pipe heavier than "standard weight." For such service conditions it has been common practice to use at least "extra strong" pipe. Paragraph *(g)* requires the use of a weight and quality at least equal to the 600-lb class of *A. S. T. M. Specification A106,* the 600-lb class being of substantially the same thickness as what has been known as "extra strong."

[A further article will cover *Sections 2* to *6* inclusive of the Code, dealing with Gas and Air Piping, Oil Piping, District Heating Piping, Fabrication Details, and Materials Specifications.—*Editor.*]

How Does Increased Steam Pressure Affect Piping Maintenance ?

One Plant's Experience . . . . By **W. H. Wilson***

OPERATING men frequently must concern themselves with effects of increased steam pressure on piping maintenance. It is felt therefore that the following description of operation with superheated steam in piping originally designed, installed and used with saturated steam for a number of years will be helpful as representing experience at one plant.

The steam-producing equipment consisted of six water-tube boilers furnishing steam at 125-lb working pressure, no superheat, steam being used for turbines, auxiliary equipment, pumps, air compressors, plant heating and process work.

In general, the main piping consisted of a 20-in. steam header, wrought-steel pipe, ⅜-in. wall thickness, Van Stone joints, with gaskets. Main valves, flanges and large fittings were semi-steel. The steam header and main supply line were supported on brackets in the boiler house, piers in tunnel and fittings with base supports in the power plant. All supports were provided with rollers to allow movement of the piping.

The piping in the power plant was reduced in size, according to the location of the branch supply lines to the power units, from 20-in. to 16-, 12-, 10- and 8-in. by using reducing flanged tees with base supports. The steam line was located near the basement floor, outlet of the tee being upward and base support on the back of the tee.

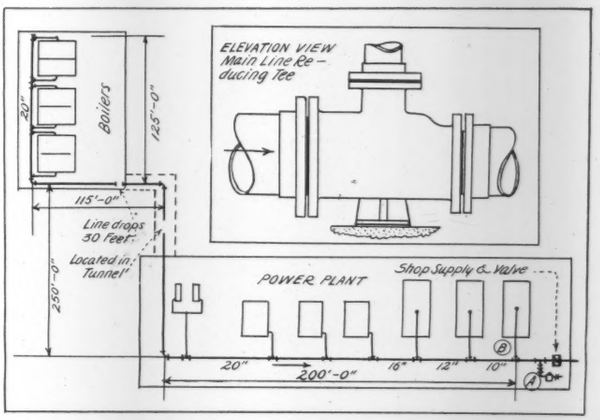
This piping was in operation for several years (at 125-lb) until the pressure was increased to 180-lb, engines, pumps, and other units being originally rated for the higher pressure. Later, after a total service of sixteen years, the boilers were equipped with superheaters and the steam temperature increased to 475 F.

The following has been observed after operating for three years with superheated steam, in comparison to the use of the same equipment previously with saturated steam.

No troubles from expansion and contraction movements of the pipe line, due to increased temperature. The travel has increased slightly, the greatest increase being observed at *B*, of approximately 1¼ in., which did not affect the branch connections to the power units. The longer section of the line in the tunnel was free to move in both directions and the increase in length was divided.

The original gaskets in the large flanged joints and valve bonnets were of the ring pattern and of a good grade of asbestos sheet packing, ⅛-in. thick; a few of these gaskets are still in service. The remainder have been renewed at later dates, some being replaced with the same type of sheet asbestos and some with corrugated monel gaskets. Both types are giving equal service with superheated steam and no real gasket trouble has been observed since the change.

*Member of Board of Consulting and Contributing Editors.



Sketch of the piping. The changes of direction allow movement due to expansion. Trap at *A*

Steam pressure is maintained on the piping continually, with the exception of periodical shut-downs for repairs, a day usually being set aside each year for this purpose.

Leaks are not readily visible with superheated steam. Valve stems, particularly small valves, small pipe connections and valve bonnets, in places not readily accessible, may leak for some time before being noticed. Frequent and careful inspections are necessary to locate steam losses from these causes and prompt repairs are essential for economy in operation.

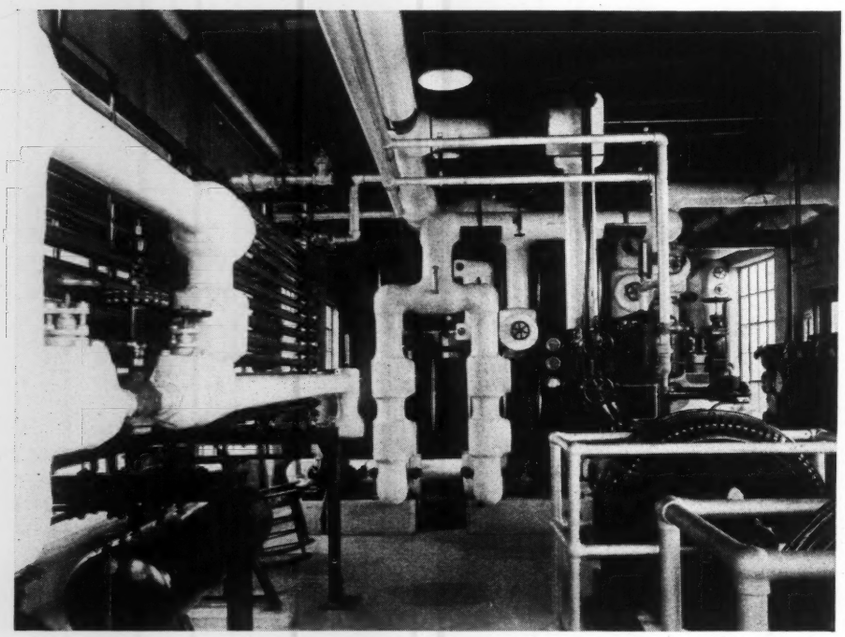
One of the first changes required to be made was in the valve-stem packing for boiler-header valves, stop valves, reducing valves and small valves on gage connections, steam auxiliary equipment, steam traps and similar connections. Valve-stem packing, usually supplied in the coil form and suitable for saturated steam, would harden or burn out in a short time when in contact with superheated steam. This was noticed especially in the valve-stem packing for governor and automatic-control valves, where it is necessary to have the valve stem move freely and change quickly with variations of the steam supply. Special packing is necessary for these valve stems.

The renewable composition discs in small valves of the globe and angle type required renewal frequently; experience has shown that particular attention should be given to all valves to see that a good grade of metal is used for the stems, seats and discs.

Observations show that pipe insulation should be considered when steam temperatures are increased. The joints in sectional pipe covering, covering loosely applied, and flanged joints not covered, while causes of lost heat and power with steam at standard temperature, are causes of increased losses when the steam temperature is increased. A complete survey of the pipe insulation is in order under these conditions.

While no direct figures are available as to cost of operation, it is the opinion of the operating force of this plant that losses from condensation are less. Repairs, maintenance and incidental costs are practically the same as before the change.

Unloaders Relieve Pressure on Pistons for Automatic Compressor Starting



By . . . . . Terry Mitchell*

A FEATURE of the refrigeration system which maintains ten different storage rooms at the Willard State Hospital, Willard, N. Y., at as many different temperatures is the unusually complete system of automatic control. The two compressors operate at 240 rpm and each delivers about 40 tons of refrigeration; they are driven by 60-hp engine-type synchronous motors supplied with 3-phase, 60-cycle, 220-volt current. Automatic starting and stopping of the motors makes necessary the use of automatic by-pass valves or unloaders to relieve the pressure on the ammonia pistons until the motors can pick up speed and become properly synchronized.

Unloaders Enable Automatic Starting and Stopping

These unloaders illustrate one of the methods available for opening the by-pass valve on an ammonia compressor by electro-mechanical means. The hook-up and operating principle are as follows:

When the temperature of the brine in the storage tank rises, the thermostat submerged in the brine completes an electric circuit which starts the exciter for one of the synchronous motors. A relay circuit meanwhile opens a pilot valve, which admits high-pressure ammonia gas to one side of a piston; the piston moves and opens the by-pass valve. The relay is made sensitive to changes of frequency in the current, induced in the fields as the main motor picks up speed; after the motor is synchronized the by-pass is allowed to close. The gas back of the piston then enters the low-pressure side of the system. Combined with what is known as direct-current latching equipment, applied to the motor starter, this unloader takes care of any momentary drops in voltage by opening the by-pass while the motor continues to stay in step; the latch holds the motor controls, for a short period, in the running position. The quick action of the gas-operated unloader is a point in its favor, in such a case.

Another type of automatic by-pass is opened and closed by a motor-and-gear operated valve, suitably wired to the

Above—The machine room at Willard State Hospital, which is maintained by the State of New York for 2,000 mentally deficient people of both sexes. This view shows one of the compressors, condensers and brine coolers

* Frick Company, Waynesboro, Pa.

thermostat and motor controls. A third design of automatic by-pass employs a valve worked by a piston, which is actuated by oil pressure instead of ammonia pressure. A small gear pump, motor driven, builds up the oil pressure in a cylinder when the valve is to be opened. When the valve is all the way up, the piston stops moving, and the oil pressure increases until it closes a contact in a pressure gage attached to the small oil cylinder. A relay from the gage starts the synchronous motor.

A fourth type of unloader consists of the recently-developed electric control valve, in which a magnetic coil operates the valve plunger directly. The plunger is inside a sealed tube; no stuffing box nor packing is required.

In each case the by-pass valve is placed between the suction inlet and discharge outlet on the compressor column, usually on the side opposite from the hand-operated valves on the manifold. To keep back the condenser pressure while the automatic by-pass is open, a check valve is installed in the discharge pipe from the compressor, and a surge drum is sometimes added to reduce pulsations.

Ammonia and Low-Brine-Pressure Cut-Outs

A magnetic stop valve, connected in the synchronous-motor circuit so as to close when the compressor stops, is fitted in the liquid feed line to each brine cooler; beyond this device is a no-freeze-back valve, to regulate the flooding of the cooler. To guard against sticking of the no-freeze-back valve, with consequent danger of freezing up the brine coolers from low suction pressure, a low-pressure ammonia cut-out, set at five lb, is arranged to stop the machine.

Hand expansion valves are provided for emergency use, and run-around connections permit removal of the automatic equipment if necessary.

If the brine circulation through the coolers should fail because the pump had stopped, the compressor would be stopped by a low-brine-pressure cut-out. A low-water-pressure cut-out keeps a similar watch over the condenser water. Should the head pressure on the machines rise excessively for any other reason, such as an accumulation of scale or non-condensable gases, a high-pressure cut-out functions to break the circuit to the main driving motor.

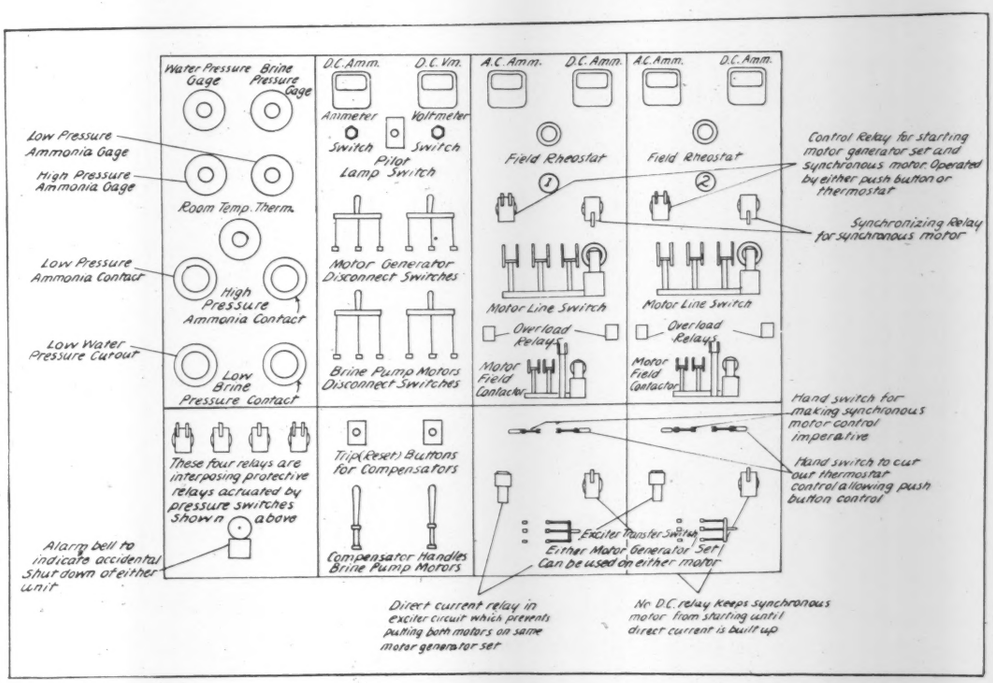

Water Pressure Gage
Brine Pressure Gage
D.C. Amm.
D.C. Vm.
A.C. Amm.
D.C. Amm.
A.C. Amm.
D.C. Amm.
Low Pressure Ammonia Gage
High Pressure Ammonia Gage
Ammeter Switch
Voltmeter Switch
Pilot Lamp Switch
Field Rheostat
Field Rheostat
Control Relay for starting motor generator set and synchronous motor. Operated by either push button or thermostat
Synchronizing Relay for synchronous motor
Room Temp. Therm.
Low Pressure Ammonia Contact
High Pressure Ammonia Contact
Motor Generator Disconnect Switches
Motor Line Switch
Motor Line Switch
Low Water Pressure Cutout
Low Brine Pressure Contact
Overload Relays
Overload Relays
Motor Field Contactor
Motor Field Contactor
Brine Pump Motors Disconnect Switches
Hand switch for making synchronous motor control imperative
These four relays are interposing protective relays actuated by pressure switches shown above
Trip (Reset) Buttons for Compensators
Hand switch to cut out thermostat control allowing push button control
Exciter Transfer Switch
Either Motor Generator Set Can be used on either motor
Alarm bell to indicate accidental shut down of either unit
Compensator Handles Brine Pump Motors
Direct current relay in exciter circuit which prevents putting both motors on same motor generator set
No D.C. relay keeps synchronous motor from starting until direct current is built up

Diagram of the control board for the two compressor units, which explains the function of the controls and switches

Size of Storage Rooms, Temperature Maintained, and Amount of 1¼-In. Wall Coils Installed

Fruit Storage	40 by 17	ft, 34 F,	970 ft pipe
Fruit Storage	42 by 12	ft, 34 F,	776 ft pipe
Fish Freezer	12 by 6	ft, 20 F,	576 ft pipe
Chill Cooler	17 by 11	ft, 24 F,	688 ft pipe
Butler Box	17 by 11	ft, 25 F,	624 ft pipe
Eggs and Cheese	17 by 11	ft, 33 F,	375 ft pipe
Cut Meats	11 by 9½	ft, 32 F,	350 ft pipe
General Meats	54 by 17½	ft, 32 F,	960 ft pipe
Salt Meats	17 by 16	ft, 38 F,	328 ft pipe
Ice Storage	20 by 10	ft, 28 F,	612 ft pipe

A by-pass in the brine mains permits short circuiting the cold solution directly to the tank from the coolers, when desired. The arrangement is such that the brine pressure on both the coils and the by-pass is controlled by a resistance valve. Separate brine strainers protect each pump.

Signal Lights Show Action of Protective Devices

Alarm and signal lights are connected to each of the protective devices, to warn in case of failure and to indicate which of the cut-outs has operated. A thermostat and alarm signal in the engineer's office also give notice if the brine temperature rises above the setting at which the compressors are supposed to start.

Thermometers are placed on both the inlet and outlet of the condensers, water jackets on the compressors, and on the brine coolers; other thermometers are located in the ammonia suction and discharge lines, outlet pipe from brine tank, and on the main return lines from the two buildings served by the system. Each cold room has its own thermometer.

Resistance valves are provided on the water overflow lines from the condensers and water jackets, as well as on the return lines from the brine coolers. There are safety relief valves on the compressors, brine coolers, and ammonia receivers, the last discharging into a diffuser on the roof of the building.

Heat Transfer *in a*

Stainless-Steel Internal-Tube Milk Heater

I AM having some difficulty with a problem and feel that you may be able to assist me. The problem is one of heat transmission in a double-pipe or internal tube heater as is used by dairies to heat milk. The heating fluid is water and its flow is counter-current to that of the milk.

Available heat transmission information as given by W. H. Motz in "Principles of Refrigeration" is as follows:

$$H = K \times A \times t_a$$

where H = total heat transfer in Btu per hr

K = heat transfer coefficient in Btu/hr/degree temp. diff.

A = area of heat transmitting surface in sq ft

t_a = mean temperature difference

K is dependent mainly upon the velocity of the fluids and is found by:

$$K = \frac{1000}{\frac{16.6}{1 + 3.33\sqrt{V_1}} + \frac{16.6}{1 + 3.33\sqrt{V_2}}}$$

t_a can be determined by higher mathematics and is reduced to

$$t_a = 0.4342 \frac{\text{g.t.d.} - \text{s.t.d.}}{\log \frac{\text{g.t.d.}}{\text{s.t.d.}}}$$

where g.t.d. = greater temp. diff. and s.t.d. = smaller temp. diff.

This formula seems to hold all right where the smaller temperature difference is not so small but as the smaller temperature difference approaches zero, as in the case of the internal-tube heaters for milk, the resultant required surface area far exceeds that actually required in practice. Actually, in the problem of the milk heaters the smaller temperature difference is zero, and thus it becomes impossible to determine the mean temperature difference by use of the formula.

An actual problem would be as follows:

Internal-tube milk heater, 8000 lb per hr of milk, using 1½-in. O. D. No. 18 stainless-steel tubes for milk and 2-in. I. D. steel tubes for water; ratio of water to milk is 10 to 1. Milk to be heated from 40 F to 143 F. Water enters at 143 F and cools to ?. Using 10-ft tubes, how many tubes required?—C. F. S., Jr.

Answer: The difficulty will be clarified if it is realized that the temperature difference *never can* be zero in an actual heat exchanger. As the temperature of the cold stream approaches that of the hot stream, the temperature difference, hence the rate of heat flow, approaches zero, and the two streams would come to the same temperature only in an apparatus of infinite length. The formula for logarithmic mean temperature difference gives a value of zero for t_a, which when used with any *finite* coefficient of heat transfer would indicate an *infinite* heating surface, as it should.

The design problem is workable only if a reasonable temperature difference is assumed at the outlet end. Assume for example that the water enters at 153 F. Since the water—milk ratio is 10 to 1, and the temperature rise of the milk is 143 — 40 = 103 F, then the temperature drop of the water will be 10.3 deg, and the water will leave at 153 — 10.3 = 142.7 F. This assumes the specific heats of water and milk to be the same. The mean temperature difference is

$$\frac{(142.7 - 40) - (153 - 143)}{2.3 \log \frac{102.7}{10}} = 39.8 \text{ F}$$

The resistance to heat transfer is best estimated by the equations proposed by McAdams in his book, "Heat Transmission." The recommended equation for the film coefficient of heat transfer to water flowing in a round pipe is

$$h = \frac{(5.6 + 0.058t)\,(G')^{0.8}}{(D')^{0.2}}$$

where h is the coefficient, Btu per hr per sq ft per F temperature difference, t is the average water temperature, F, G' represents the mass velocity of the water, lb per sec per sq ft of cross-section, and D' is the diameter in inches.

Assuming that this equation may be used as an estimation of the coefficient for milk, the procedure is as follows:

$$t = 40 + \frac{103}{2} = 91 \text{ F for milk}$$

$$t = 153 - \frac{10.3}{2} = 148 \text{ F for water}$$

$D' = 1.50$ in. for milk

$D' = 2.07 - 1.58 = 0.49$ in. for the annular space carrying water, since for an annular space the equivalent diameter of a round pipe is twice the clearance between the tubes. The outer 2 in. pipe is assumed to have an inside diameter of 2.07 inches.

The cross-section of the 1.5-in. pipe is 1.76 sq in., and that of the annular space is

$$\frac{3.14}{4}(2.07^2 - 1.58^2) = 1.40 \text{ sq in.}$$

Consequently the mass velocity G' is

$$\frac{8000}{3600} \times \frac{144}{1.76} = 182 \text{ lb per sec per sq ft for milk, and}$$

$$\frac{8000 \times 10}{3600} \times \frac{144}{1.40} = 2280 \text{ lb per sec per sq ft for water}$$

Substituting these values in the above equation there results $h = 633$ for milk, and $h = 7900$ for water. Assuming the stainless steel to have a thermal conductivity of 14 Btu/(hr) (sq ft) (F/ft) and a wall thickness of 0.04 inches, the thermal resistance of the metal wall, per foot of length is

$$r_t = \text{tube wall resistance} = \frac{0.04}{12} \times \frac{12}{14 \times 3.14 \times 1.54} = 0.00059$$

The water-side resistance, per foot of length, is

$$r_w = \frac{1}{7900} \times \frac{12}{3.14 \times 1.58} = 0.00031$$

The milk-side resistance, per foot of length, is

$$r_m = \frac{1}{633} \times \frac{12}{3.14 \times 1.5} = 0.0040$$

The over-all thermal resistance is the sum of these three,

$$R = 0.00059 + 0.00031 + 0.0040 = 0.0049$$

The total heat transfer is

$$8000 \times (143 - 40) = 824{,}000 \text{ Btu per hr.}$$

The required tube length is consequently

$$\frac{824{,}000}{39.8} \times 0.0049 = 102 \text{ ft, or 11 tubes.}$$

The given water—milk ratio of 10 to 1 leads to an abnormally high water velocity of about 37 ft per sec in the annular space. This high water rate would involve excessive pressure drop on the water side, and inefficient use of hot water. The milk velocity is quite reasonable.

The equation used is based on data in clean tubes. Suitable factors of safety must be introduced if there is likelihood of scum or other deposits on the heat-transfer surfaces. It should be emphasized also that the calculations given are based on the problem as stated, modified to the extent of assuming the water to enter at 153 F. The calculations may be repeated for other inlet water temperatures to determine the effect on the required tube length.—T. K. SHERWOOD.*

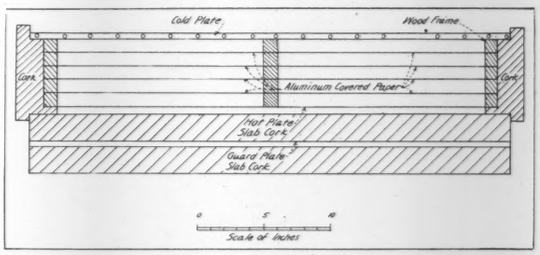
*Assistant Professor of Chemical Engineering, Massachusetts Institute of Technology, Cambridge, Mass.

Metallic Insulation

By **A. F. Dufton***

THE surfaces at controlled temperatures which will surround the experimental room shortly to be erected at the Building Research Station, Garston, need to be well insulated on the outside, both to insure accuracy of temperature control and also from the point of view of economy of operating costs. A new insulating material, consisting of stout paper or similar material covered on both sides with aluminum foil, has recently become commercially available, and the possibility of using such insulation is contemplated. The material is cheap and likely to be effective, but information as to its insulating value is at present scanty. It was decided, therefore, to measure the transmission of heat through a specimen typical of the structure to be erected.

It appeared that an equivalent of three inches of cork would be satisfactory and a specimen was accordingly made up to afford approximately this degree of insulation.



Test set-up for determining thermal conductivity of metallic insulation

The thermal conductivity measurements were made at the National Physical Laboratory, Teddington, and the following description is based upon the report received from the Laboratory:

The object of the experiments was to determine the rate of heat transfer through a structure consisting of two plane surfaces with four partitions in the space between.

The surfaces measured 32 in. by 32 in. and were 5 in. apart. The four sheets of aluminum paper were separated from each other by 1-in. air-spaces as shown in the sketch.

The hot plate of the conductivity apparatus was maintained at a temperature of approximately 90 F and the cold plate at 68 F. The surfaces of both plates were blackened.

The aluminum-covered paper was glued on wooden frames of 1-in. section, the frames being of the same external dimensions as the hot plate. Down the center of each frame a wooden bar of 1-in. section was fixed in order to support the paper.

The frames and the hot and cold plates were clamped together and insulated on the sides with slab cork.

The hot plate was also backed with slab cork and with a "guard plate" maintained at the same temperature as the hot plate, to eliminate heat leakage from the back of the hot plate. The cold plate was maintained at a steady temperature by water circulation through a coil attached to the back of the plate and temperatures were measured at three points on each of the hot and the cold plates.

The heat transmitted by the specimen was obtained from a measurement of the watts dissipated in the re-

Position of Surfaces	Temperature of Hot Surface F	Temperature of Cold Surface F	Thermal Conductivity in Btu per sq ft per hr per F difference in temperature between surfaces
(*a*) Surfaces vertical Hot || Cold	88	66	0.13
(*b*) Surfaces horizontal, hot surface uppermost Hot Cold	95	68	0.10
(*c*) Surfaces horizontal, cold surface uppermost Cold Hot	90	63	0.11
(*d*) Surfaces at 45° to vertical, hot surface uppermost Hot Cold	91	68	0.11
(*e*) Surfaces at 45° to vertical, cold surface uppermost Cold Hot	90	68	0.13

*Chief Physicist, Building Research Station, Garston, England. Published by permission from the September, 1933, *Journal of The Institution of Heating & Ventilating Engineers.*

sistor of the hot plate after correction for the heat leakage through the cork lagging around the sides of the structure.

Tests were made with the surfaces—

(*a*) vertical,
(*b*) horizontal, hot surface uppermost,
(*c*) horizontal, cold surface uppermost,
(*d*) at 45 deg to vertical, hot surface uppermost.
(*e*) at 45 deg to vertical, cold surface uppermost.

The results given in the table have been corrected for the heat conducted through the wooden frames.

These results are published in the hope that they will be of service to others who may contemplate the use of such insulation. The small variation in conductivity produced by change in inclination of the specimen is noteworthy. The insulating value of three inches of slab cork is 0.11 Btu per sq ft per hr per F difference in temperature between faces.

For rough calculations it is convenient to reckon a closed air-space, not less than ¾ in. wide and bounded on one or both sides by a plane bright metallic surface, as affording insulation equivalent to ½ in. of cork.

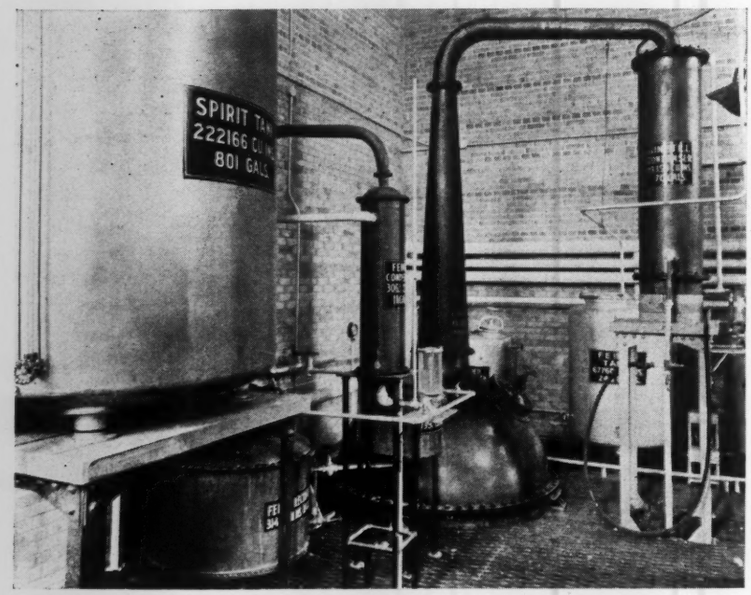
Mine Modernizes Pump Control—Saving Exceeds $10,000 per Year*

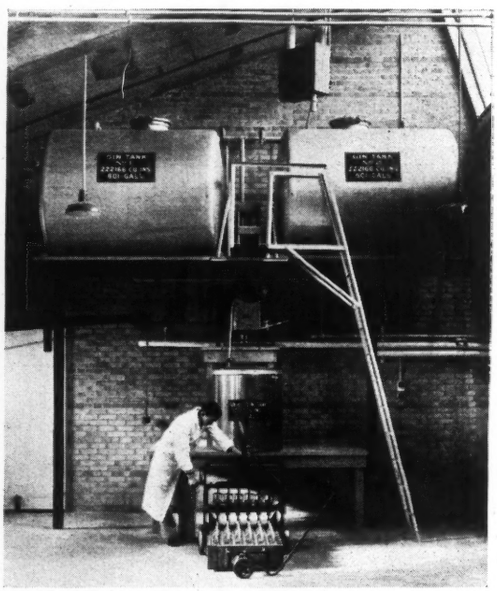
IN operating a copper mine it was necessary to pump out approximately 1200 gallons of water per minute to prevent the mine becoming flooded. There were three levels in the mine from which water was pumped, namely the 1200-ft level, the 1400-ft level, and the 1800-ft level. Duplicate pumping equipment was installed at each level so that if for any reason one failed to operate, or was shut down for adjustment or repairs, the other pump would be available for use to prevent the mine becoming flooded. The two centrifugal pumps at the 1200-ft level were 600 hp each; at 1400-ft level, 100 hp each, and at the 1800-ft level, 300 hp each. Manual control was used throughout.

It was felt that the replacement of manual control by automatic control would remove the hazard of the mines being flooded due to human error in operating the existing control equipment and would thus justify the expenditure of approximately $8,000. The installation was made with suitable float switches to control the operation of the automatic equipment. Suitable signals were installed so that in case trouble with the pumping system did occur anywhere within the mines, the operator would be notified.

As a result, direct operating expenses were reduced $30 per day—somewhat over $10,000 a year. In addition appreciable savings in maintenance on the pump and control equipment was effected, but since this account was not segregated this saving could not be definitely evaluated. Thus the $10,000 figure is conservative as the realized savings were considerably more than that.

*Information courtesy of W. B. Clark, General Electric Company.





Photos courtesy The International Nickel Company, Inc.

NICKEL PIPING IN CANADIAN GIN PLANT

Left—Corner of the distillery in new gin plant of W. and A. Gilbey Ltd., New Toronto, Ont., Canada, showing 801-gallon spirit tank made of 88-12 nickel-chrome. There is a small nickel tank behind the gin flavoring still used for the storing of *feints*. All of the piping is pure nickel. Right—The bottling room, showing two 801-gallon gin tanks used to charge the 100-gallon bottling tanks below. Storage tanks are 88-12 nickel-chrome, and pure nickel is used for the bottling tank

Heating·Piping
and Air Conditioning

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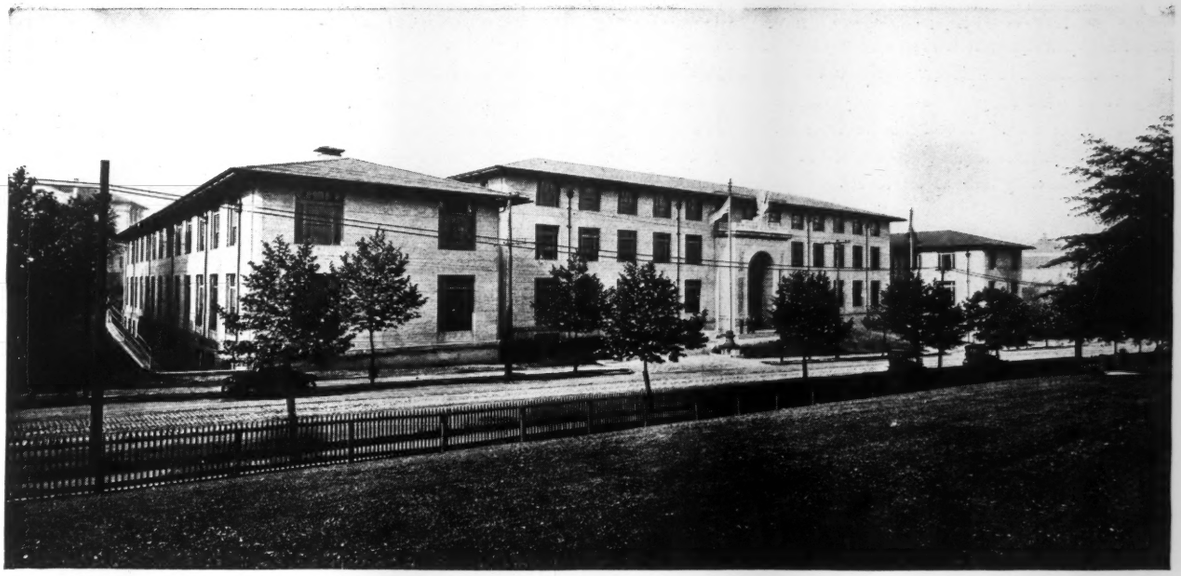

Pittsburgh Experiment Station of the U. S. Bureau of Mines where the Research Laboratory of the American Society of Heating and Ventilating Engineers is located

Studies of Solar Radiation Through Bare and Shaded Windows

By **F. C. Houghten*, Carl Gutberlet**, and J. L. Blackshaw*** *(MEMBERS)*, Pittsburgh, Pa.**

FORMER studies[1] made at the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS have indicated that solar radiation through windows is a large portion of the cooling load in summer air conditioning. A study,[2] made by Walker, Sanford, and Wells in co-operation with the Society, of the sources of heat comprising the summer cooling load in air conditioning a modern office building has shown solar radiation through windows to be the predominating factor. Consideration of these results by the Committee on Research led to their authorization of the investigation here reported concerning heat gain through windows from solar radiation under different weather conditions, and as affected by several types and applications of window shading appurtenances.

The investigation was made by the A. S. H. V. E. Research Laboratory in Pittsburgh in a two-room test house, Fig. 1 and Fig. 2, built for the purpose on a high, open plot of ground unaffected by shadows. The construction, consisting of board insulation on the outside and inside of 2 by 4-in. studding with a blanket insulation fill, had a wall transmittance of 0.06 Btu per square foot per hour per degree temperature difference from air to air. All joints were caulked to give a low infiltration rate. The east and west walls and the roof were shaded with aluminum painted canvas in order to minimize the

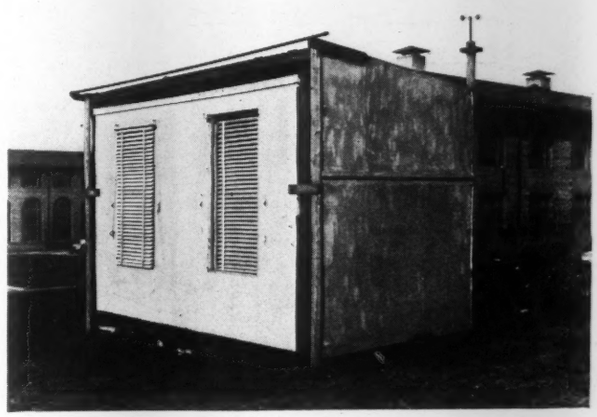

Fig. 1—Test House with Venetian Blind Applied to Inside Window of East Room (right) and to Outside Window of West Room

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[1]Absorption of Solar Radiation in Its Relation to the Temperature, Color, Angle and Other Characteristics of the Absorbing Surface, by F. C. Houghten, and Carl Gutberlet. A. S. H. V. E. TRANSACTIONS, Vol. 36, 1930, p. 137.

Heat Transmission as Influenced by Heat Capacity and Solar Radiation, by F. C. Houghten, J. L. Blackshaw, E. M. Pugh, and Paul McDermott. A. S. H. V. E. TRANSACTIONS, Vol. 38, 1932, p. 231.

[2]Field Studies of Office Building Cooling, by J. H. Walker, S. S. Sanford, and E. P. Wells. A. S. H. V. E. TRANSACTIONS, Vol. 38, 1932, p. 285.

For presentation at the 40th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, New York, N. Y., February, 1934.

effect of solar radiation. The inside walls, ceilings, and floors were painted gray. Dark gray cheesecloth screens were placed within the rooms so as to intercept all direct radiation from the sun through the windows and keep it from being absorbed by the floors, the walls, or the cooling units. Each room had a window facing due south made of a single sheet of double strength A-quality window glass with a clear unobstructed area of 17.24 sq ft.

Each room contained a commercial ice cooling unit, a small electric heater, a psychrometer with an air circulating fan and with thermocouples to indicate wet- and dry-bulb temperatures, a thermostat, an air circulating fan, two Nicholls heat flow meters, and necessary wiring and piping connections. The temperature in each room was maintained by balancing with the thermostat the cooling unit against the electric heater. The rate of cooling was controlled by automatic intermittent operation of the air circulating fan within the unit and by hand adjustment from outside the rooms of inlet and outlet dampers.

Thermocouples selectively connected to a potentiometer set-up made it possible to observe (1) air temperatures at various points within the rooms, (2) surface temperatures of inside and outside walls, ceilings, floors, and window glass, (3) outside air temperatures a few inches from the glass, (4) outside air temperatures in the shade, and (5) surface temperatures at desired places on the various window appurtenances used. Nicholls heat flow meters applied to the centers of the inside east wall of the east room and of the west wall of the west room determined the heat flow through the surfaces of these walls. Heat flows through other surfaces of side walls, floors, and ceilings were obtained by a second movable meter in each room.

The rate of cooling supplied to each room was determined by collecting the water produced by the melting ice in calibrated graduates at 15-minute intervals and observing the temperature of this water at the point where it left the rooms. The heat input by the fan motors and electric heaters was obtained from periodic reading of the indicating watthour meters for each room.

Solar intensity was determined by the Laboratory's pyrheliometer, which was directed continuously toward the sun by a clock and trunnion mounting. Wind velocities were obtained from an electrically recording cup anemometer located above the roof.

Consideration was given to the most desirable temperature to be maintained within the test rooms. At first, tests were made with varying room temperatures which would hold the temperature differential between the inside and outside to that advocated by THE A. S. H. V. E. GUIDE 1933[3] for summer air conditioning. It was soon found, however, that this so complicated the problem that computation would not give satisfactory results, so it was decided to select that temperature which would give the recommended differential at noon and to hold it constant during respective tests. To insure equilibrium conditions of the structure and air during a test, an attempt was made to anticipate this differential the evening before the test and to maintain this room temperature during the night. However, if morning weather conditions indicated that a poor estimate had been made, the air temperature was changed in the direction desired before the sun became effective or the test started, but this change was never greater than 3 deg in either direction. The ice charge in the cooling units so controlled the relative humidity that the dew-point in the room was always in the neighborhood of 40 F.

In making a test, ice sufficient to last at least 24 hours was put in the coolers the previous evening, the required adjustments to the window appurtenances and to the temperature control equipment inside the rooms were made, and the rooms were sealed. Before the sun became effective on the south wall of the building in the morning, the operation of the equipment was checked, the necessary adjustments were made, and the test was started. The test was considered to be ended at any time after the sun ceased to be effective on the south wall or when cooling was no longer required to maintain the desired temperature.

Shades tested were of the ordinary commercial spring

[3]Table 2, Chapter 22, A. S. H. V. E. GUIDE, 1933.

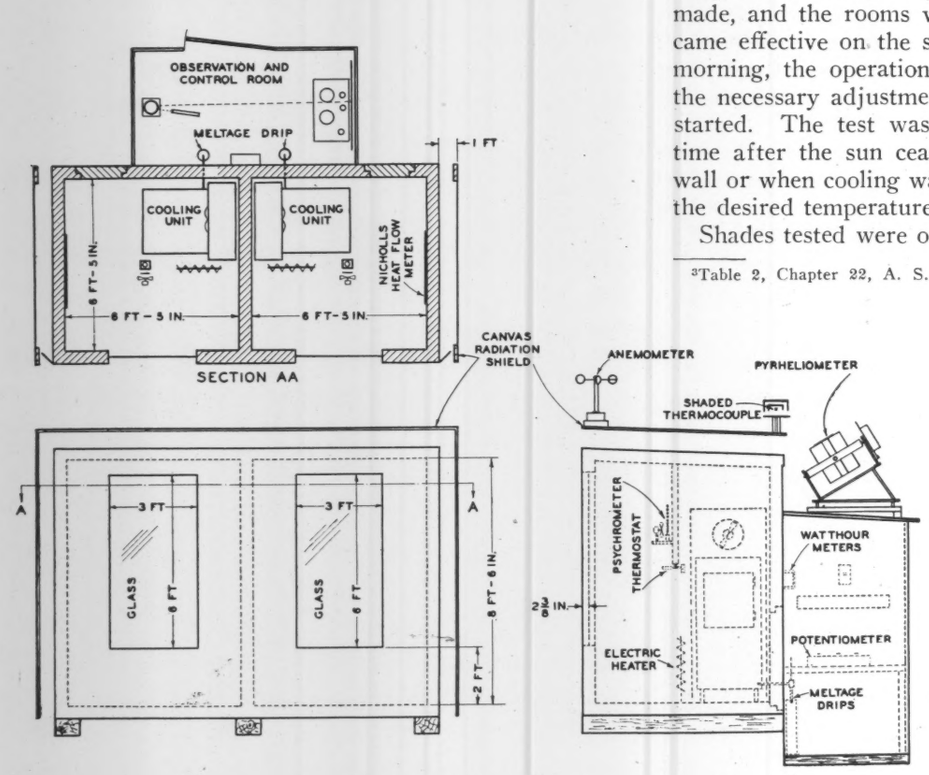

Fig. 2—Test House

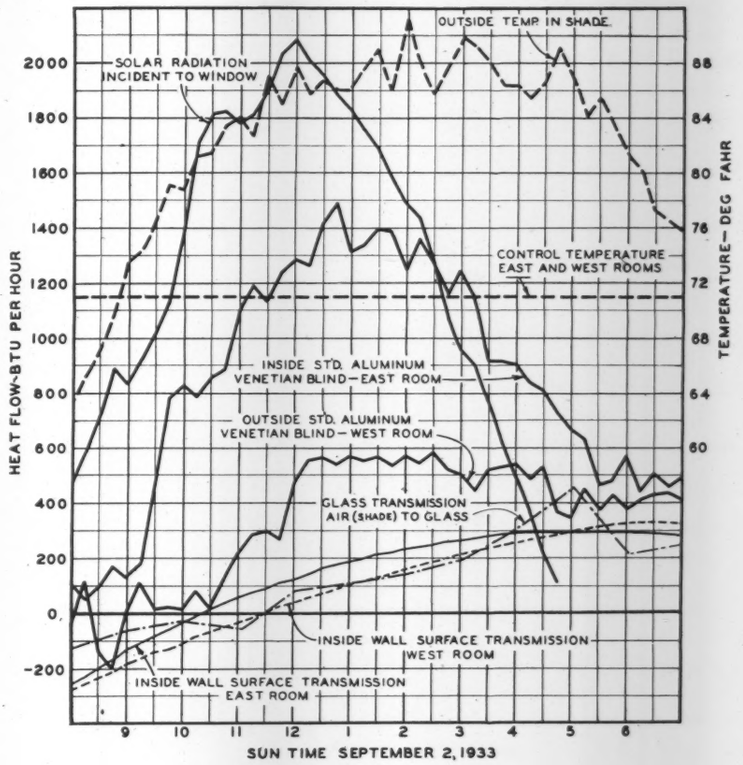
OUTSIDE TEMP IN SHADE
SOLAR RADIATION INCIDENT TO WINDOW
CONTROL TEMPERATURE EAST AND WEST ROOMS
INSIDE STD. ALUMINUM VENETIAN BLIND—EAST ROOM
OUTSIDE STD. ALUMINUM VENETIAN BLIND—WEST ROOM
GLASS TRANSMISSION AIR (SHADE) TO GLASS
INSIDE WALL SURFACE TRANSMISSION WEST ROOM
INSIDE WALL SURFACE TRANSMISSION EAST ROOM
HEAT FLOW-BTU PER HOUR
TEMPERATURE—DEG FAHR
SUN TIME SEPTEMBER 2, 1933

Fig. 3—Heat Transfer and Temperature Relationships for Inside and Outside Venetian Blinds

roll type made of a filled cotton cloth of medium weight. Their original finish was a buff color, but for some tests they were painted on the side facing the sun with metallic aluminum in lacquer. In their standard application they overlapped the window frame by approximately 1 in. all around and hung quite snugly to it. For a few tests, wooden cleats were nailed around the shade to hold it so tightly to the frame that there was a minimum transfer of convected air; while for one inside test, the shade was hung with an air gap of approximately 1½ in. between its periphery and the frame of the window to give free circulation of convected air. For another inside test, the shade was half drawn.

Commercial Venetian blinds with two different finishes were tested. Some were stained a dark green; and others were finished by sputtering molten aluminum metal over the wooden slats to affix a solid metallic aluminum coating which both reflected and diffused solar energy falling upon it. The slats of all Venetian blinds tested were 2⅜ in. wide, ⅛ in. thick, spaced 1¾ in. apart. A Venetian blind was considered to have a standard adjustment when it hung close to the frame of the window, fully extended, with its slats set at an angle of 45 deg so they excluded the sun's rays. For one test the slats were closed tightly, and for another they were wide open in a horizontal position. Venetian blinds intended for use on the outside were designed with features which enabled them to be hung away from the window, as an awning without sides, at any adjustable angle up to about 30 deg. Several tests were made with such a blind hung as an awning at the maximum angle, but one test was made with the angle adjusted during the day to admit a maximum amount of light without any direct sunshine getting through the sides.

The awnings used in the study were of a commercial type having sides and an 8-in. decorative fringe, made of medium weight striped canvas supported on a steel framework. In their standard application they projected 44 in. from the building and left the tops of the windows at angles of 40 deg. The tops and sides of the awnings were tightly fastened to the walls of the house. For most tests, the awnings were used just as purchased (standard finish), but for one test an awning was given a heavy coat of metallic aluminum in lacquer on its outside surface.

Complete shading was accomplished by four large canvas shields placed in front of the window to shade the opening from solar radiation. The nearest canvas was held about 7 in. from the glass and 5 in. from the wall, and the other three shields were placed successively farther away and were separated by three 4-in. air spaces. This arrangement allowed free air circulation between the canvas and the window and between the different layers of canvas which eliminated both direct radiation from the sun and reradiation from the canvas nearest the glass. Hence, the only heat transfer through the window should have been that due to conduction from the stratum of shaded air nearest the window.

Test Results and Analyses

Twenty-five successful tests were made on different window arrangements, with results as listed in Table 1. Five tests, listed also in Table 1, were satisfactory in one room only because of unforeseen difficulties which affected the other. Many additional tests were started but discontinued because of unsatisfactory weather conditions.

The results of Test 32 are plotted in Fig. 3. This test was made on September 2, with an aluminum Venetian blind hung inside the window in the east room and a similar blind hung outside the window in the west room. A control temperature of 71 F had been chosen for each room because the outside temperature in the morning was low, but its rapid rise to a high of 91 F at 2:00 p. m. proved this to have been a poor choice, for 78 F would have been in better accord with THE GUIDE recommendation for inside temperature for such conditions. The cooling rate, as shown for each room, reached a maximum of 1480 Btu per hour for the east room, which had the inside blind, and 580 Btu per hour for the west room, which had the outside blind. The negative values occurred when the electrical input required to maintain the room temperature was greater than the heat extracted by the cooler.

Heat flow through the walls, ceilings, and floors of the two rooms calculated from the heat meter readings, and heat transfer from the outside air to the outside glass surface for the prevailing temperature difference and wind velocity for the east window equipped with the inside blind are also plotted in Fig. 3. The great lag in

Table 1—Analysis of Cooling Load with Bare and Shaded Windows

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38
				Solar Radiation Incident to Glass Btu per Hour						East Room														West Room														
										Appurtenance				Rates Btu per Hour										Appurtenance				Rates Btu per Hour									
														Cooling					Transmission	Heat Removal								Cooling					Transmission	Heat Removal			
Test Number	Date ~ 1933	Max. Outside Temp.	Max Solar Radiation ~ Normal to Sun's Rays ~ Btu per Sq Ft per Hour	Max. ~ Entire Window †	Max. per Sq Ft of Glass	Av. ~ Entire Window *	Av. per Sq Ft of Glass *	Inside Control Temp.	Av. Air Temp. Difference Inside to Outside (in shade) *	Type	Finish	Location	Adjustment	Max. ~ Entire Room	Max. ~ Window Only	Max. ~ per Sq Ft of Glass	Av. ~ Entire Room (9 AM to 5 PM)	Av. ~ Entire Room *	Av ~ Entire Wall *	Av ~ Outside Air (shade) * to Glass Bare Window Assumed	Av. Charged to Window * (14)-(20)	Av. Charged to A * Sq Ft of Glass	(19)-(20) ÷ 100 (7)+(21) Per Cent Transmission by Appurtenance	Type	Finish	Location	Adjustment	Max. ~ Entire Room	Max. ~ Window Only	Max. ~ per Sq Ft of Glass	Av. ~ Entire Room (9 AM to 5 PM)	Av. ~ Entire Room *	Av. ~ Entire Wall *	Av. Outside Air (shade) * to Glass Bare Window Assumed	Av Charged to Window * (33)-(34)	Av. Charged to A * Sq Ft of Glass	(33)-(34) ÷100 (7)+(35) Per Cent Transmission by Appurtenance
4	July 13	86.7	287	1441	83.6	1072	62.2	75.8	6.3	Complete Shading				428	370	15.7	234	234	178	42	36	3.2	5	Bare Window				1361	1366	79.2	1027	1027	-66	42	1094	63.4	98
6	17	82.4	302	1572	91.2	1029	59.7	76.0	1.3	Shade	Buff	Inside	Cleated	657	489	28.4	419	448	149	-123	299	17.3	33	"	"			1076	1182	68.6	696	744	-153	-123	897	52.0	99
7	18	86.3	224	1104	64.0	677	39.3	76.4	4.4	"	"	"	With Gap	751	693	40.2	514	504	91	109	413	24.0	53														
8	19	92.3	259	1374	79.7	946	54.9	76.4	10.4	"	"	"	St'd	943	521	30.2	719	732	187	127	545	31.6	51	Bare Window				1531	1408	81.7	1112	1173	101	127	1072	62.2	100
9	20	95.6	247	1443	83.7	1024	59.4	76.4	10.8	"	Al.	"	"	1177	733	42.5	825	850	390	109	460	.267	41	"	"			2045	1784	103.5	1229	1310	202	109	1108	64.3	98
10	21	99.7	254	1385	80.3	970	56.3	76.4	17.4	"	"	"	Cleated	972	652	37.8	719	734	276	285	458	26.6	36	"	"			1868	1643	95.3	1341	1409	207	285	1202	69.7	96
12	23	102.6	252	1386	80.4	977	56.7	76.4	20.9	"	"	Outside	"	584	170	9.9	546	528	306	381	222	12.9	16	"	"			1910	1658	96.2	1459	1512	215	381	1297	75.2	96
13	24	95.4	238	1306	75.7	837	48.6	76.4	13.0	"	Buff	"	St'd	789	554	32.1	529	571	216	273	355	20.6	32	"	"			1780	1560	90.5	1138	1275	151	273	1124	65.2	101
15	31	97.8	247	1484	86.1	1026	59.5	79.4	14.3	"	"	Inside	Cleated	858	595	34.5	729	742	253	262	489	28.4	38	Shade	Buff	Inside	½ Drawn	1219	1022	59.3	1022	1039	170	262	869	50.4	68
16	Aug. 1	97.2	243	1483	86.0	991	57.5	813	10.8	"	"	"	St'd	947	610	35.4	784	829	312	165	517	30.0	45														
17	4	80.5	311	1940	112.5	1360	78.9	813	-7.5															Awning	St'd.	Outside	St'd.	297	447	25.9	128	121	-169	-223	290	16.8	26
18	7	89.0	309	2020	117.2	1342	77.8	80.4	3.4	Awning	St'd	Outside	St'd.	534	384	22.3	398	403	95	-40	308	17.9	24	"	"	"	"	525	521	30.2	334	330	-21	-40	351	20.4	27
20	11	87.9	297	1887	109.5	1283	74.4	76.4	5.5	"	Al.	"	"	515	392	22.7	375	370	79	20	291	16.9	22	"	"	"	"	629	587	34.0	494	456	22	20	434	25.2	33
21	12	87.9	252	1713	99.4	1243	72.1	77.3	7.0	Ven. Bl.	Green	"	As Awn.	559	434	25.2	481	488	93	92	395	22.9	30	Bare Window				1775	1696	98.4	1147	1192	-28	92	1220	70.8	91
22	15	86.1	281	1994	115.7	1305	75.7	75.3	5.5	"	"	"	"	607	481	27.9	404	410	44	20	366	21.2	28	Complete Shading				196	186	10.8	78	49	-39	20	88	5.1	7
23	16	89.4	277	1963	113.9	1249	72.4	74.6	7.0	Bare Window				1591	1506	87.4	1066	1102	42	-134	1060	61.5	95														
25	18	85.6	309	2123	123.1	1129	65.5	None	——	"	"								No Cooling					Bare Window				No Cooling									
26	21	84.9	262	1979	114.8	1389	80.6	76.4	4.4	Ven. Bl.	Al.	Outside	Adjusted	465	383	22.2	356	337	51	-10	286	16.6	21	Complete Shading				349	322	18.7	235	236	166	-10	60	3.5	4
27	22	80.4	301	2149	124.6	1281	74.3	74.6	1.0	"	"	Inside	St'd.	1009	921	53.4	771	771	66	-106	705	40.9	60	"	"			120	130	7.5	87	87	27	-106	60	3.5	5
28	25	90.7	283	2180	126.5	1484	86.1	79.0	6.3	"	"	"	"	1115	988	57.3	819	832	89	-105	743	43.1	54	Bare Window				1784	1810	105.0	1222	1255	-21	-105	1276	74.0	93
31	30	82.8	286	2344	136.3	1432	83.1	73.0	4.6	Shade	Buff	Outside	"	641	577	33.5	429	436	48	-5	388	22.5	27	Shade	Al.	Outside	Cleated	332	294	17.1	235	236	6	-5	230	13.3	16
32	Sept. 2	91.2	245	2093	121.4	1359	78.8	71.0	14.0	Ven. Bl.	Al.	Inside	"	1388	1212	70.3	1019	1030	141	127	889	51.6	60	Ven. Bl.	"	"	St'd.	559	465	27.0	374	381	77	127	304	17.6	20
33	5	87.5	206	1806	104.8	1064	61.7	71.0	10.5															"	"	"	"	349	299	17.3	263	263	46	158	217	12.6	18
34	6	90.9	252	2273	131.8	1610	93.4	74.0	11.4	Ven. Bl.	Al.	Outside	St'd.	535	364	21.1	369	369	113	182	256	14.8	14	"	"	"	As Awn.	623	508	29.5	541	541	32	182	509	29.5	28
35	7	92.1	235	2111	122.4	1414	82.0	80.0	6.1	"	"	"	"	374	270	15.7	202	202	24	40	178	10.3	12	"	"	"	Wide Open	548	462	26.8	375	375	0	40	375	21.8	26
36	8	94.5	227	2052	119.0	1394	80.9	83.0	6.7	"	"	"	"	454	319	18.5	268	268	74	52	194	11.3	13	"	"	"	Closed	198	168	9.7	122	122	11	52	111	6.4	8
37	9	90.3	212	1969	114.2	1115	64.7	83.0	3.3	"	"	"	"	312	239	13.9	237	237	55	-40	182	10.6	17	"	Green	"	St'd.	283	276	16.0	183	183	36	-40	147	8.5	14
38	10	81.9	272	2559	148.4	1501	87.1	79.0	-0.5	"	"	"	"	360	266	15.4	276	276	80	-148	196	11.4	15	"	"	"	"	306	328	19.0	189	189	-13	-148	202	11.7	15
39	17	83.2	265	2670	154.9	1416	111.1	79.0	2.5	Bare Window				2467	2502	145.1	1619	1619	29	-221	1648	95.6	97	Complete Shading				138	75	4.4	-87	-87	-159	-221	72	4.2	4.
41	Oct 11	71.8	280	3582	207.8	2408	139.7	57.9	9.8	Shade	Buff	Inside	St'd.	1617	1490	86.4	1163	1163	75	138	1088	63.1	43	Shade	Al.	Inside	St'd.	1411	1401	81.3	1081	1081	-13	138	1094	63.5	43
Av. All Tests		88.8	265	1913	111.0	1261	73.1	76.5	7.1										130	54														30	54			

*For period—9 a. m. until the sun ceased to be effective.
†Window glass area—17.24 sq ft.

Abbreviations

With Gap—1½ in. between periphery of shade and frame.
St'd—Standard.
Plain canvas awning.
Shade typically applied.
Venetian blind—hanging down—slats at 45 deg.
Al.—Aluminum.
Ven. Bl.—Venetian blind.
As Awn.—Venetian blind adjusted away from window, as awning without sides.
Adjusted—Venetian blind used as awning but moved periodically to keep sun off window.
Wide open—Venetian blind—slats horizontal.
Closed—Venetian blind—slats vertical.

flow through the insulated walls is indicated by the continuance of negative values after the outside air temperature became higher than that inside. Wall transfer to the east room was higher than that to the west room during the early part of the day because the air between the shading canvas and the wall was at a higher temperature on this side of the building during the forenoon.

The maximum rate of solar radiation impingement against the 17.24 sq ft of window area, as determined from the pyrheliometer reading, is 2,093 Btu per hour at noon and falls off rapidly before and after this time because the angle of impingement lessens as the altitude and azimuth angles of the sun change and because part of the glass area is shaded by the frame. The rate of energy impingement on the total area of glass at any instant is given by the formula:

$$E = P\left(\sin\left[\cos^{-1}\sqrt{\{\cos\alpha\cos(90^\circ-\Upsilon+\Theta)\}^2+\sin^2\alpha}\right]\right)\left(wh-hL\tan(\Upsilon-\Theta)-wL\tan\alpha+L^2\tan(\Upsilon-\Theta)\tan\alpha\right)$$

where:

E = solar intensity, in Btu per hour, on the window, h feet high, w feet wide, set back a distance L feet from the outside wall surface.

P = solar intensity, in Btu per square foot per hour, normal to the direction of radiation.

α = altitude angle of the sun at the instant considered.

Υ = angle between north and the direction the window faces.

Θ = azimuth angle of sun (angle between sun and the north).

The average rate at which energy impinges against the entire glass, given in Table 1 for the period of the day considered, was obtained by integrating the instantaneous rates. This average rate divided by the total glass area gave the average rate per hour per square foot of glass.

The sun became effective on the south window at 7:05 a. m. on September 2 and disappeared at 4:45 p. m. This time of exposure changed from day to day during the summer, being a period of from 8:20 a. m. to 3:40 p. m. on July 15, and one from 6:40 a. m. to 5:20 p. m. on September 15. For the purpose of making a heat balance, the periods from 9:00 a. m. until the sun ceased to be effective were considered, but some values are listed for a longer period, running from 9:00 a. m. to 5:00 p. m., which includes the late afternoon hours when cooling was required because the high air to air temperature difference continued and because there was a lag in heat flow through the walls.

The average rate of radiation impingement against the south glass, 17.24 sq ft in area, for the period from 9:00 a. m. to the time the sun ceased to be effective on September 2, as given in Table 1, was 1,359 Btu per hour. The effect of altitude and azimuth angles of the sun and of shading by the frame in reducing the solar radiation on the glass is illustrated by comparing this value with 121.4 Btu per square foot (2,093 Btu for 17.24 sq ft) maximum solar intensity, and with 245 Btu per square foot (4,263 Btu for 17.24 sq ft) average solar intensity normal to the direction of radiation.

In analyzing the results of the tests to show the relation of the cooling load to different window conditions and to show the effectiveness of window appurtenances, it was necessary to consider all sources of heat entering and leaving the test rooms. These sources of heat are illustrated in Fig. 4. Solar radiation A of a given intensity impinges against the glass surface at an angle

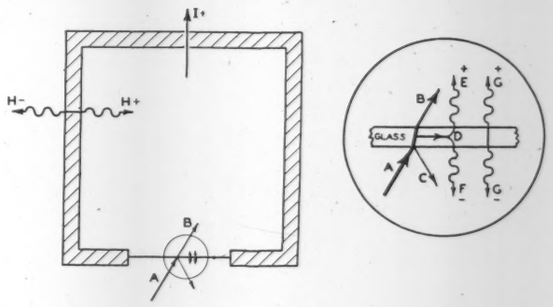

Fig. 4—Analysis of Heat Gain

which changes throughout the day. A part B of the radiant energy passes directly through the glass into the room. Another portion of it C is reflected from the glass surface, while the remaining portion D is absorbed while passing through the glass. The energy D raises the glass temperature so that heat E is transferred from the inside glass surface to the air inside the room, or heat F is transferred from the outside glass surface to the air outside the room, or both. The division of D into E and F is dependent upon the relative temperatures of the glass and of the air on either side. Simultaneously, heat transfer G is taking place through the glass between the outside and inside air, its direction and magnitude depending upon the temperature difference. Table 2 shows the temperature gradients obtained in several tests of bare windows, and cites conditions when absorption of radiation in the glass was sufficient to raise its temperature above both the inside and outside air temperature in the shade, or when heat flowed from both surfaces of the window at the same time.

The solar radiation A, Fig. 4, incident to the glass plus the conduction F between the outside air and the glass surface gives the total heat gain through the window, excepting for the effect of direct reflection of radiation C from the outside glass surface. A brief study made of the absorption and reflection of solar radiation by glass, which will be presented later in this report, indicates that the reflection C is negligibly small. Therefore, in the analysis of the data presented in this paper reflection C is disregarded.

The magnitude and direction of heat flow H through the walls of the room are dependent upon the temperature difference between the air on the inside and that on the outside, but since there is considerable lag in the transfer through a wall, owing to its heat capacity and other factors, the rate of transfer between the inside air and inside surface of the wall and between the outside air and outside surface of the wall may differ widely in

Table 2—Temperature Gradients at Noon from Outside Air in Shade Through Glass to Inside Air With and Without Shading

Test	Date	Temp. in Shade F	Room 1			Room 2		
			Appurtenance	Glass Temp. F	Control Temp. F	Appurtenance	Glass Temp. F	Control Temp. F
4	July 13	81.8	Complete Shading	78.2	75.9	Bare Window	84.2	75.9
27	Aug. 22	82.7	Inside Ven. Blind	84.6	74.7	Complete Shading	76.7	74.7
39	Sept. 17	80.5	Bare Window	85.4	79.0	Complete Shading	80.0	79.0

magnitude or even in direction at any instant, and may bear little or no relation to the air to air temperature difference. *I* represents the heat removal from the room occasioned by melting ice.

In order to maintain a constant air temperature within the room, the rate of heat removal *I* should equal the summation of all the sources of heat gain. Analysis of the results supported this but showed a small lag; that is, the rate of heat removal from the room always lagged somewhat behind the heat gain. This is demonstrated by the heat removal curves, Fig. 3, which continue after the sun ceased to be effective on the window at a rate greater than the combined rates of heat transfer through the inside wall surface and the glass.

It was desired in analyzing the results of the different tests listed in Table 1 to equate the heat entering through the windows and the heat removal chargeable to the windows in order to establish a balance and account for all transfers.

The calculated heat gain through the window, which is the sum of the radiation impingement on the glass, Column 7, Table 1, and the transfer from the outside air to the glass surface, Columns 21 or 35 (in the absence of a shading appurtenance), is compared with the heat removal charged to the window as determined by the total meltage corrected for electric input, Columns 19 or 33, less the calculated transfer through the walls, Columns 20 or 34. The ratio of the actual heat removal for the respective rooms to the heat which would be gained through an unshaded window is expressed in per cent in Columns 24 and 38.

Averages for all tests made with the same window conditions are given in Table 3, where five tests with the windows completely shaded have an average percentage ratio of bare window transmission of 5 per cent. This seems a reasonable figure.

Table 3—Average Values for Window Conditions

Appurtenance†				Number of Tests Averaged	Average Rates of Cooling All Tests Btu per Hour							Per Cent Transmission by Appurtenance	Greatest Variation of Any Test from the Average Per Cent Transmission
Type	Finish	Location	Adjustment		Max. ~ Entire Room	Max. ~ Window Only	Max. ~ per Sq Ft of Glass	Av. ~ Entire Room (9 AM to 5 PM)	Av. ~ Entire Room*	Av. Charged to Window*	Av. Charged to A* Sq Ft of Glass		
Complete Shading				5	246	217	11.4	109	104	63	3.9	5	2
Ven. Bl.	Al.	Outside	Closed	1	198	168	9.7	122	122	111	6.4	8	
"	Al+Green	"	Std.	9	359	314	18.2	262	263	208	12.1	15	5
Shade	Al.	"	Cleated	2	458	232	13.5	390	382	226	13.1	16	0
Ven. Bl.	"	"	Adjusted	1	465	383	22.2	356	337	286	16.6	21	
Awning	"	"	Std.	1	515	392	22.7	375	370	291	16.9	22	
Ven Bl.	"	"	Wide Open	1	548	462	26.8	375	375	375	21.8	26	
Awning	Std.	"	Std.	4	496	485	28.1	339	328	346	20.1	28	5
Ven. Bl.	Al+Green	"	As Awn.	3	596	474	27.5	475	480	423	24.5	29	2
Shade	Buff	"	Std.	2	715	565	32.8	479	503	371	21.5	30	3
"	Al+Buff	Inside	Cleated	3	829	579	33.6	622	641	415	24.1	36	3
"	"	"	Std.	5	1219	951	55.2	914	931	741	43.0	45	6
"	Buff	"	With Gap	1	751	693	40.2	514	504	413	24.0	53	
Ven. Bl.	Al.	"	Std.	3	1171	1040	60.3	870	878	779	45.2	58	4
Shade	Buff	"	½ Drawn	1	1219	1022	59.3	1022	1039	869	50.4	68	
Bare Window				11	1744	1647	95.5	1187	1238	1182	68.5	97	6

*For period—9 a. m. until the sun ceased to be effective.
†For abbreviations, see Table 1.

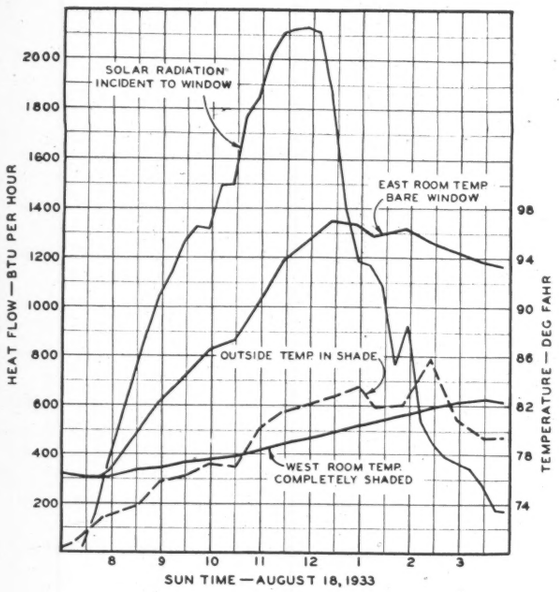

Fig. 5—Relation of Inside Room Temperatures to Bare and Completely Shaded Windows—Rooms Not Conditioned

The percentages given in Tables 1 and 3 for bare windows represent for eleven tests the heat removed by cooling which was chargeable to the window in relation to that calculated to enter the window. These tests show an average percentage of 97, with a maximum variation of 6 per cent for any test. The discrepancy between these percentages and 100 per cent are chargeable to other sources of heat gain, or to errors in temperature or wind observations on which conductances were based. Probably the largest sources of error involved are the inability to include in the analysis reflection of radiation at the glass surface, and the acceptance of film transfer coefficients for the prevailing wind and temperature conditions used in calculating the heat gain through the windows by conduction.

Similar percentage ratios of bare window transmission are given in Tables 1 and 3 for the windows when affected by various types, finishes, locations, and adjustments of window appurtenances. These percentages seem rather conclusive, and their averages should be acceptable to ± 5 per cent where several tests were made with the same window condition. This tolerance does not permit drawing close comparisons between appurtenances of their conditions of application when they show little variation. However, when such tests were made on the same day a direct comparison between the two rooms should be more representative of the actual relationship of the appurtenances.

Temperature Rise in Unconditioned Rooms

A test was made on August 18 to determine what temperatures would be obtained in the two test rooms on a bright day when no

cooling was supplied to either room. The window in the east room was left bare, and that in the west room was completely shaded. Fig. 5 shows that at the start of the day each room had a temperature of 76 F, and that while the outside temperature rose to 86 F, that of the unshaded room reached a maximum of 97 F while that in the shaded room was only 82.5 F. It can be seen from the curves that the temperature peak in the shaded room lagged behind the peak in the unshaded room by several hours. Although no quantitative results were obtained from this test, the comparative temperatures found illustrate the desirability of applying shades to the outsides of windows.

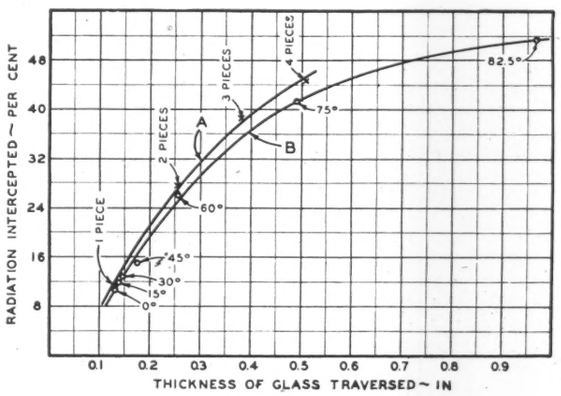

Fig. 6—Solar Radiation Intercepted by Various Thickness of Double Strength A-Quality Window Glass

Curve *A*—Various Pieces Normal to Direction of Radiation
Curve *B*—One Piece at Various Angles of Incidence with Radiation

Interception of Solar Radiation by Glass

During the summer, a related study was made of the comparative percentages of solar radiation intercepted, either by absorption or reflection, by different kinds and thicknesses of glass held at various angles of incidence with the sun's rays. Table 4 gives the percentage of energy intercepted for the different conditions studied. Curve *A* in Fig. 6 shows the relation between the percentage of radiation intercepted and the thickness of the glass, when one or more pieces of double strength window glass of the same quality as that in the test house were held normal to the sun. Curve *B* in Fig. 6 shows this percentage relation when a single piece of such glass was held at varying angles of incidence with the sun's rays.

No account was taken in this analysis of the refraction of the rays as they passed through the glass. It will be noted that the relation between the radiation intercepted and the thickness of the glass traversed is approximately the same whether the thickness results from placing several pieces of glass normal to the rays, or from changing the angle of incidence of a single piece. This indicates that the increased interception occasioned by changing the angle of incidence results from increased absorption within the glass because of its greater thickness rather than from the reflection, and that reflection, shown as *C* in Fig. 4, is not an important factor in heat transfer through a window by solar radiation. This assumption was used in the analysis of the data. It should be pointed out, however, that the decreased transmission occasioned by the piling up of several pieces of glass normal to the sun's rays may have resulted from partial reflections at the several surfaces, rather than from increased thickness. It would be of interest to make a more thorough study of the effect had on solar radiation transmission through glass by the thickness of the glass and the angle of incidence, but neither facilities nor time were available for such an extended investigation of this phase of the subject.

Air Motion in the Test Rooms

In order to maintain uniform air velocities in the rooms at all times independent of the rate of cooling required and in order to keep such velocities within practical limits, an auxiliary fan and housing was located above each of the commercial ice cooling units. Air supplied through the cooling unit by the fan contained therein was discharged into this housing intermittently as the thermostat turned the cooler fan on and off, and at different rates as the inlet and outlet dampers to the units were adjusted. Since the rate of cooling required was always low in comparison with the capacity of the units used, the rate of air flow through the unit was always small. A by-pass around the cooling unit took air from the floor level and admitted it to the housing where it and the cooler air combined before entering the auxiliary fan which mixed and blew the air horizontally into the room at the 6-ft level. The free delivery capacity of the auxiliary fan was much greater than its capacity to draw air through the by-pass in parallel with the cooling unit. Hence, the air was discharged to the room over a considerable area at a relatively low velocity.

The maximum velocity 30 in. in front of the fan and 3 ft from the wall toward which the air was directed for maximum rate of discharge through the unit was

Table 4—Solar Radiation Intercepted by Glass of Various Types, Thicknesses, and at Various Angles

Type of Glass	Thickness			Angle of Incidence Deg	Radiation Intercepted Per Cent
	Pieces	Each Piece In.	Traversed by Radiation In.		
Plate: Yellow tinge	1	0.255	0.255	0	14.8
Plate: Green tinge	1	0.240	0.240	0	16.6
Plate: Clear	1	0.270	0.270	0	12.8
Window: Single strength; A-quality	1	0.071	0.071	0	10.7
Photographic	1	0.070	0.070	0	9.9
Window: Double strength; A-quality	1	0.127	0.127	0	11.2
Window: Double strength; A-quality	2	0.127	0.254	0	27.8
Window: Double strength; A-quality	3	0.127	0.381	0	38.4
Window: Double strength; A-quality	4	0.127	0.508	0	45.7
Window: Double strength; A-quality	1	0.127	0.127	0	10.8
Window: Double strength; A-quality	1	0.127	0.131	15	11.9
Window: Double strength; A-quality	1	0.127	0.147	30	12.7
Window: Double strength; A-quality	1	0.127	0.180	45	14.9
Window: Double strength; A-quality	1	0.127	0.254	60	26.0
Window: Double strength; A-quality	1	0.127	0.490	75	41.6
Window. Double strength; A-quality	1	0.127	0.969	82.5	53.7

209 fpm. The maximum velocity as determined by the Kata thermometer 10 in. from the wall toward which the air was directed was 83 fpm. Complete surveys of the rooms indicated few velocities elsewhere as high as 60 fpm, while at most points near the walls the velocity was found to be less than 40 fpm. Since the Society's ventilation standards approve velocities up to 50 fpm for a well ventilated room, those velocities pertaining in the test rooms were not excessive and should not adversely affect the rate of transfer through any of the walls or windows.

Infiltration

Special attention was given to the design and construction of the test rooms to eliminate, as far as possible, the heat gain by infiltration. Every joint in the building, both in the framework and in the insulating board covering all walls, was well caulked. Each thickness of blanket insulation filling the studding spaces was self-contained in tarred paper. It was flanged tightly to the top, bottom, and sides of the studding, with all joints sealed with roofing cement. The entire inside and outside surfaces of the house were painted with three coats of good quality lead paint, care being taken to fill all cracks. The glass was sealed into place with ½-in. moulding strips embedded in putty in the window frames, which were an integral part of the wall construction. The small openings for entrance into the rooms were made of specially flanged construction and were always sealed by rubber gaskets and caulking compound during tests.

Carbon dioxide tests, made in accordance with methods used by Armspach[4] in earlier Laboratory studies, gave infiltration rates of not to exceed 18 cu ft per hour for each of the rooms. For the maximum inside and outside temperature differences and the dew-points experienced in the tests, this rate of infiltration would cause a combined sensible and latent heat gain to the rooms of not to exceed 0.5 per cent of the average total rate of heat removal for all tests with bare windows. Tests were made to insure that infiltration did not increase above this value with age of the structure, but no account was taken of this small heat gain in the computations.

Practical Application of the Results

While this report gives laboratory results on windows with southern exposure only, the fact that the results obtained in calculating the heat gain to the rooms check closely with the measured rates of cooling makes it possible to use this method for calculating the heat gain for windows of any other exposure. The factors entering such calculation are the rate of solar radiation, the direction of exposure, and the glass conduction as determined by the inside and outside temperature difference and the wind velocity. The percentage by which this rate of heat gain is reduced by the application of any of the shading appurtenances here reported may be applied to such calculated heat gain through any bare window. Practical values of heat gain through windows and appurtenances under conditions of different exposure and for the weather conditions met with in Pittsburgh during the summer of 1933 are intended to be the subject of another paper.

The average maximum rates and the average of the daily average rates of heat gain through the windows, given in Table 3 for the conditions studied during the summer of 1933, may be used in design. The average per cent of the solar radiation through a bare window which is transmitted through the same window when a given shading appurtenance is used may be taken from Table 3 as a direct measure of the usefulness of such shading device.

[4]Infiltration of Air in Buildings, by O. W. Armspach, A. S. H. V. E. TRANSACTIONS, Vol. 27, 1921, p. 121.

Summary and Conclusions

1. An analysis of the sources of heat gain through a window is made. Calculated heat gains are compared with measured rates of heat removal for bare windows, for completely shaded windows, and for windows with several types of shading appurtenances applied in different ways.

2. Energy transfers through shaded windows are compared with the energy transferred through a bare window. Results showed that 5 per cent of the solar radiation through a bare window was transferred through a completely shaded window, 22 per cent through an outside Venetian blind, 28 per cent through an awning, 45 per cent through an inside shade, and 58 per cent through an inside Venetian blind. Similar percentage transfers are also given for other variations in both kind of finish and application of these appurtenances.

3. Heat removal rates actually required to cool the rooms with the various window appurtenances and the rates of heat removal charged only to the windows are given.

Acknowledgment

The authors acknowledge the valuable assistance rendered by Paul F. McDermott, until recently a member of the Laboratory staff but now associated with the Johns-Manville Research Laboratories.

Ontario

December 4, 1933: The regular meeting was held at the Royal York Hotel and, following dinner, Secy. H. R. Roth read the minutes of the two preceding meetings. Thomas McDonald, chairman of the Membership Committee, introduced two new members who were present, O. L. Maddux and E. T. Whittall.

Pres. W. P. Boddington drew attention to the 40th Annual Meeting of the Society to be held in New York in February and announced that H. S. Moore would serve the Chapter as representative and F. E. Ellis as alternate on the Nominating Committee, which will meet at that time.

President Boddington told of requests for a social gathering and, after a discussion, it was voted that H. H. Angus act as chairman of a committee including R. W. McHenry and one other member to report at the next meeting.

Mr. McHenry proposed that the Society sponsor an organization to establish a code for figuring heating jobs, similar to the Red Seal Electric League now operating in Toronto. Following an extended discussion, the members decided that while the matter was of interest it was not in the function of the A. S. H. V. E. to engage in a commercial enterprise.

President Boddington introduced the principal speaker of the evening, A. S. Leitch, of Toronto, who spoke on The Development of Heating and Ventilating Within the Last 30 Years.

Observations of Hay-Fever Sufferers in Air-Conditioned Room and the Relationship Between the Pollen Content of Outdoor Air and Weather Conditions

By T. A. Kendall* and Garland Weidner†, M. D., C. P. H. *(NON-MEMBERS)*
Lexington, Ky.

IN the spring of 1932 a room known as Dicker Hall, located in the mechanical engineering building on the University of Kentucky campus, was equipped with a unit air-conditioner. Dicker Hall is the social club room for the students of the College of Engineering and is normally used for that purpose only, but at the beginning of the hay fever season of 1932, through announcements in local newspapers, the room was made available to the public as offering a possible haven for hay fever sufferers. The invitation met with a hearty response. No records of the visitors were obtained during this season but many visitors reported informally the relief of some of their symptoms.

At about the same time, due to interest in air filters for filtering pollinated air, a study of the pollen content of outdoor air was begun. Slides coated with corn oil were exposed daily outdoors in front of the engineering building (Station A). Metal slide shelters were used similar to those described by Gay.[1] The slides were exposed each morning between 8 and 9 o'clock and were allowed to remain for a period of 24 hours.

In making the counts, a Bausch and Lomb microscope, having a standard 16 mm objective and a 10X ocular, was used. An area of 113 sq mm was observed by three trips across the slide. These counts were multiplied by a factor (0.885) giving the number of pollen per square centimeter. A plot of the daily counts for the 1932 season is shown in Fig. 1.

So far as making any practical use of the 1932 counts, the principal interest settled around the relationship between pollen counts and weather conditions. Meterological data were obtained from the local weather bureau and were tabulated for each day as shown in Table 1. Plots of precipitation in inches, percentage of sunshine and relative humidity are shown in Fig. 1.

In 1926, Ray M. Balyeat, M.D., made a study of the pollen content of the air and weather conditions in Oklahoma City.[2] The relationship between precipitation and pollen count is stated by Dr. Balyeat as follows:

> The amount of precipitation plays a very definite part in several ways in determining the amount of pollen that actually remains in the air. In the first place, the amount and distribution of rainfall during the growth of plants determine largely the number of plants and the extent of their growth. They also determine the amount of pollen that will be produced by the plant life, inasmuch as large stalks of Bermuda or ragweed will produce more pollen than small ones.

After the period of anthesis arrives, Dr. Balyeat apparently considers the washing of pollen from the air as the only effect of precipitation upon the pollen content of the air. On the day following a heavy rain in Oklahoma City, the pollen count reached one of the highest peaks of the season. Dr. Balyeat says:

> On the day following the heavy precipitation first mentioned, the wind began to blow about 3 o'clock in the morning and continued throughout the day. The percentage of sunshine on the following day was 100. Under such conditions the precipitation had practically nothing to do with regulating the amount of pollen in the air.

It is the opinion of the writers however, that this high count was due to the effect of precipitation as well as sunshine upon plant life, for the reason that a rain causes the plants to grow more rapidly and produce an increasing amount of pollen to be scattered during the dry period following the rain. For example, the high counts obtained on August 28 and September 2, 14 and 19, as shown in Fig. 1, occurred on days preceded by rainfall. This was true for September 2, in spite of the fact that the percentage of total possible sunshine was very low on this day. The total precipitation shown in Fig. 1 was for the same period covered by the pollen count but the precipitation curve does not show the part of the 24-hour period during which the precipitation occurred; the high precipitation and high pollen count for some days appear to occur simultaneously, and hence the foregoing statement does not seem to be substantiated by the data as plotted. This can be explained by reference, for instance, to the complete data for September 2. The precipitation on this day fell between 11:00 a.m. and 2:00 p.m. and was followed by bright sunshine; while the pollen count obtained for the same date was for the 24-hour period ending about 9 o'clock the following morning.

Dr. Balyeat states that a high percentage of sunshine causes a high count. The pollen counts of August 29,

The research work reported on in this paper was conducted in the Johnston Solar Laboratory of the College of Engineering, University of Kentucky, under the general direction of Dean F. Paul Anderson and in cooperation with the University of Kentucky Department of Hygiene.

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†Department of Hygiene, University of Kentucky.

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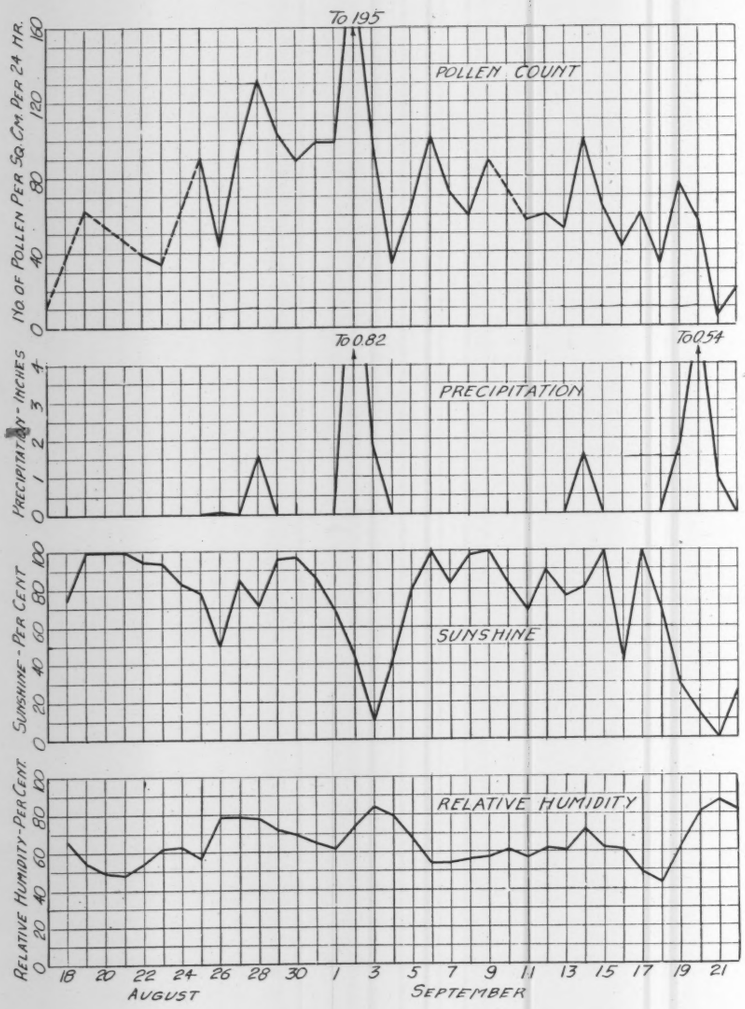
To 195
POLLEN COUNT
No. of Pollen Per Sq. Cm. Per 24 Hr.
0 40 80 120 160
To 0.82
To 0.54
PRECIPITATION
Precipitation - Inches
0 1 2 3 4
SUNSHINE
Sunshine - Per Cent
0 20 40 60 80 100
RELATIVE HUMIDITY
Relative Humidity - Per Cent
0 20 40 60 80 100
18 20 22 24 26 28 30 1 3 5 7 9 11 13 15 17 19 21
August
September

Fig. 1—Daily Pollen Count and Weather Conditions During the Ragweed Season, Lexington, Ky.—1932

30, and 31, and on September 6 were probably above normal due to the high percentage of sunshine on those days. For a day or so following a peak in the pollen count, the count is usually low, regardless of weather conditions. This is due probably to a shortage of matured anthers caused by the previous heavy yield. For example, the low count on September 4 may have been caused by the heavy yield which occurred on September 2.

Relative to wind movement, Dr. Balyeat says:

> The velocity of wind, to a considerable extent, determines the amount of pollen that gets into the air, how far it will be carried, and the amount that comes in contact with the nasal mucous membrane of the hay fever sufferer. The time of day in which the wind blows is important. For example, if there is a stiff gale throughout the early morning hours, during which time the pollination is the heaviest, it will carry more pollen into the air than a similar gale during the evening.[3]

The average velocity in Lexington during the 1932 season was 9.89 miles per hour. There were five days with an average velocity above 12 miles per hour and three days with a velocity below 8 miles per hour. A study of the data has failed to reveal any correlation between wind velocity, direction of wind and pollen count. The effect of wind was probably outweighed by other weather influences. Also it is probable that no correlation between these factors could have been obtained even though other weather conditions had remained constant, for the reason that the sources of the pollen were fairly well distributed around the station.

The relationship between the pollen content of the air and weather conditions may be summarized by stating the following general conclusions which appear to be substantiated by the data:

> (1) The effect of rain is twofold: (a) rain washes the air practically free of pollen; and (b) rain accelerates the growth of the plant and causes a high yield of pollen following the rain. (2) A high percentage of sunshine will increase the pollen yield due to the ripening and opening of matured anthers. (3) Following a peak in the pollen count the count is usually low for a day or so, regardless of weather conditions, because of a shortage of matured anthers due to the previous heavy yield. The magnitude of the count will, of course, vary with the seasonal trend, being relatively low at the beginning and end of the season and relatively high during the middle of the season.

When the 1933 hay fever season approached it was decided to make Dicker Hall available again for hay fever sufferers; and this time, to observe the visitors and make a record of the relief they obtained. Also, the pollen count of the outdoor air was continued, and in addition a record of the pollen content of the air in Dicker Hall was made, supplemented by a record of the room temperature and relative humidity. Outside weather conditions were observed also, but due to a curtailment of the activities of the weather bureau complete meterological data were not obtained. Another outdoor station (Station B), located at the residence of one of the writers, about a mile and one-half from the university campus, was established to obtain a check count of the pollen content of the air and to observe the difference due to location. Table 2 shows the indoor and outdoor pollen counts, temperatures and humidities, and precipitation for each day of the 1933 season. A plot of the pollen counts for Stations A and B, and for the air conditioned room is shown in Fig. 2.

Comparing the outdoor counts for the 1932 and 1933 seasons, it will be noticed that the distribution for the 1932 season was fairly uniform from August 19 to September 20, while the count for 1933 shows a decided upward trend until the peak was reached near the middle of the season and then a more precipitous but yet gradual and fairly even decline to the minimum at the end of the season. This difference in the distribution of pollen for the two seasons may be accounted for by the fact that during the 1933 season practically all of the precipitation came within one period of seven days, from August 28 to September 3, while in 1932 there were four separate rains occurring at intervals during the season with nearly clear days between. It will be noticed that the count at Station B was much higher than at Station A during the 1933 season. This was probably caused

Table 1—Tabulation of Data—1932

Date	No. of Pollen Per Sq Cm Per 24 Hr	Dry Bulb Temp. F	Relative Humidity Per Cent	Sunshine Per Cent	Precipitation Inches	Average Wind Velocity Miles per Hour							
						North	North-East	East	South-East	South	South-West	West	North-West
Aug.													
17	11	....	....	....	....	....	....	....	....	....	....	....	....
18	....	70.1	66	75	T	6.16	.83	....	....	....	....	.50	3.91
19	63	66.5	54	100	....	3.75	4.25	4.04	....	....	....	....	....
20	....	67.9	49	100	....	....	5.04	8.66	....	....	....	....	....
21	....	71.3	48	100	....	....	1.75	10.25	....	....	....	....	....
22	39	72.9	54	95	....	1.25	.46	8.25	....	....	....	....	....
23	34	74.6	62	94	....	....	.58	2.26	2.83	2.21	....	....	....
24	....	75.4	63	83	....	....	....	....	....	5.25	3.92	.67	....
25	91	80.6	57	78	....	....	....	....	....	5.79	5.66	.96	....
26	43	73.0	79	50	.01	....	....	....	....	....	11.09	2.00	....
27	95	76.5	79	85	....	....	.13	.25	.25	....	.83	1.92	3.92
28	132	76.6	78	71	.16	....	....	....	.38	5.75	1.75	1.17	.46
29	103	80.0	72	96	T	....	....	....	....	.25	3.96	5.00	....
30	89	80.2	69	97	T	.38	....	....	....	....	1.88	2.08	.83
31	99	82.8	65	86	....	....	5.41	.79	1.79	.96	....	....	....
Sept.													
1	99	83.5	62	69	T	....	1.17	3.46	3.62	2.00	....	....	....
2	195	79.4	74	45	.82	2.33	.25	4.66	.92	....	.92	....	....
3	96	74.7	84	10	.18	....	....	....	1.46	4.42	5.87	3.92	....
4	34	75.5	79	43	....	1.50	.25	.21	....	....	4.67	1.21	.25
5	63	72.9	67	81	....	4.42	....	.13	....	....	....	....	4.75
6	102	64.1	54	100	....	9.34	2.79	....	....	....	....	....	....
7	72	61.4	54	83	....	5.38	6.70	.38	....	....	....	....	....
8	60	63.9	56	98	....	4.16	4.54	4.34	....	....	....	....	....
9	89	70.6	57	100	....	....	....	10.91	....	.88	....	....	....
10	....	72.8	61	83	T	....	1.08	4.17	2.38	1.75	....	....	....
11	57	73.6	57	68	....	.25	1.96	3.21	2.34	.29	....	.50	.79
12	60	73.0	62	90	....	....	....	1.08	1.50	2.54	4.96	....	....
13	52	73.9	61	76	....	.29	1.00	.54	.88	1.46	2.21	2.04	....
14	100	71.5	72	81	.16	1.58	1.67	2.75	.25	.21	.25	.33	.21
15	64	72.6	62	100	....	6.21	.88	....	....	....	....	....	1.38
16	42	66.0	61	41	....	1.54	....	....	....	....	....	.54	6.66
17	60	64.6	48	100	....	....	....	....	2.88	.46	.21	....	4.84
18	33	73.8	43	69	....	....	....	....	6.54	7.87	....	....	....
19	76	73.5	63	28	.18	....	....	2.29	.46	6.25	3.00	....	....
20	56	67.1	81	13	.54	....	....	2.62	4.59	3.71	4.29	.38	....
21	5	66.2	87	0	.09	....	....	.63	1.92	1.25	.25	2.21	1.71
22	19	69.3	81	25	T	6.75	.50	.92	....	.50	....	....	....
23	1	61.8	69	29	....	3.12	10.21	....	....	....	....	....	....
24	24	59.4	57	100	....	.54	10.18	3.12	....	....	....	....	....
25	19	65.2	55	62	.09	....	....	9.00	4.00	1.00	....	....	....
26	4	63.9	87	0	.60	....	....	....	....	12.39	.50	....	....
27	4	64.7	86	6	.10	.88	....	....	....	1.75	7.38	....	3.21
28	2	56.9	65	98	....	2.00	....	....	....	....	....	1.13	6.30
29	3	58.6	56	55	....	2.46	3.71	3.12	.79	....	....	.58	1.25
30	0	59.0	60	97	....	....	....	.87	.37	.46	.25	.25	3.20

Table 2—Tabulation of Data—1933

Date	Number of Pollen per Sq Cm per 24 Hrs			Temperature and Humidity						Precipitation	
				Outside 24 Hrs.		Outside 9 AM to 9 PM			Air Conditioned Room 9 AM to 9 PM			
	Station A	Station B	Air Cond. Room	D. B. T. F	% Rel. Hum.	D. B. T. F	Eff. T. F	% Rel. Hum.	D. B. T. F	Eff. T. F	% Rel. Hum.	Inches	Time of Day
Aug.													
15	13	....	....	71.5	62	75.0	68.6	53	69.8	66.0	66	....	....
16	5	9	....	74.0	67	78.9	72.6	58	72.5	69.0	72	....	....
17	8	....	0	71.0	83	74.0	71.0	79	74.2	70.5	73	.10	Noon
18	37	77	2	74.6	71	80.0	75.2	67	74.4	70.4	70	T	....
19	45	146	0	73.1	67	77.7	71.7	59	74.9	70.9	70	....	....
20	29	81	0	75.0	67	79.2	72.8	58	74.8	70.9	70	....	....
21	31	....	2	76.0	73	81.0	75.6	67	75.6	71.6	71	....	....
22	80	151	5	74.4	71	78.0	72.7	64	74.8	70.7	70	....	....
23	97	240	4	75.6	68	81.0	74.1	58	75.0	71.0	71	....	....
24	47	109	5	73.8	74	77.7	72.6	67	74.5	70.9	74	....	....
25	73	110	8	76.6	68	81.6	74.8	59	76.6	72.8	72	....	....
26	77	326	3	79.7	61	84.0	75.8	52	75.9	71.5	69	....	....
27	119	322	0	79.0	65	85.0	76.1	49	76.6	72.5	71	.01	Last hour
28	165	174	2	67.4	84	67.7	66.5	90	72.1	69.4	76	.35	First morning
29	173	196	13	70.8	65	74.1	68.3	57	72.0	68.6	74	.02	Last morning.
30	47	172	11	69.8	84	69.8	67.5	80	71.0	68.5	78	.57	Both mornings
31	75	182	2	71.7	84	74.6	72.0	81	75.5	72.5	78	.05	First morning
Sept.													
1	161	368	1	74.5	78	75.2	71.6	75	76.1	72.8	76	....	....
2	129	516	4	75.1	86	76.5	74.0	83	75.8	72.9	79	.12	Afternoon
3	71	233	0	74.1	88	77.3	74.8	83	74.6	72.0	81	1.74	Evening
4	182	304	0	74.5	90	75.9	74.5	88	76.5	74.0	82	....	....
5	62	378	4	75.8	76	81.2	76.1	69	77.0	73.9	77	....	....
6	212	361	2	75.9	71	80.5	74.5	64	77.4	73.5	73	....	....
7	156	304	0	78.0	72	82.7	76.3	63	76.3	72.4	72	....	....
8	124	189	2	79.7	76	84.1	78.3	67	75.5	71.9	73	....	....
9	133	318	3	81.2	72	85.0	78.9	67	76.6	72.7	73	....	....
10	121	....	1	79.4	71	82.7	76.8	64	76.4	72.2	70	T	....
11	71	*22	6 1	79.4	74	85.0	78.3	62	76.9	72.9	72	.58	Last morning
12	27	274	3	75.0	85	78.3	75.2	80	74.6	71.4	76	.56	During day
13	27	126	0	75.0	86	76.7	74.1	83	74.9	71.5	75	.65	During night
14	24	41	4	76.4	85	80.5	76.6	77	75.1	71.6	75	.29	Afternoon
15	17	60	0	75.0	86	78.1	75.1	80	76.5	73.1	76	....	....
16	31	84	0	78.2	80	80.4	76.3	75	76.6	73.2	76	....	....
17	17	44	2	75.3	60	80.9	73.6	54	77.4	72.9	70	....	....
18	21	23	3	70.6	61	75.3	68.6	53	77.3	72.0	64	....	....
19	12	50	0	77.8	63	82.2	76.0	61	76.5	72.1	69	....	....
20	9	39	0	64.5	48	70.1	63.6	42	73.7	68.7	61	....	....

*One half of count for 48 hours.

by the source of the pollen being nearer Station B than Station A, and perhaps also to some relative difference in the degree of exposure of the slides.

The unit air-conditioner installed in Dicker Hall was of conventional design as shown in Fig. 3. It consisted of a three wheel fan with separate outlets, a heater, and two banks of sprays. The return air entered near the top of the unit at the rear, traveled downward parallel to one spray, then upward on the front side counter to the other spray, through the heater, fan and discharge outlets. The water for the sprays was cooled by means of refrigeration equipment already installed in connection with the Johnston Solar Laboratory which is adjacent to Dicker Hall.

The control of the unit was entirely by hand, through the regulation of the quantity of spray water used, the manipulation of a mixing valve which regulated the relative amount of refrigerated water and recirculated water, and through the adjustment of the fan speed by means of a four-step electric controller. The heater was not used at any time.

There were a few days when the outside temperature was relatively low, and it was necessary to run with a relatively high spray water temperature to avoid cooling the room below a comfortable temperature, and sometimes to cut off the spray water entirely to

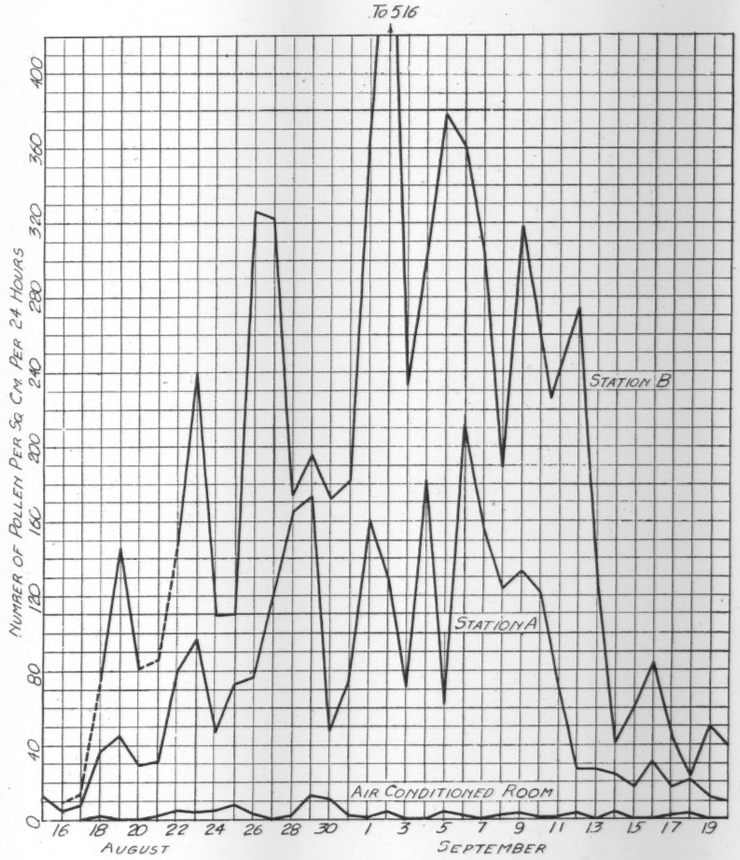

Fig. 2—Daily Pollen Count and Weather Conditions During the Ragweed Season, Lexington, Ky.—1933

avoid increasing the relative humidity. To provide for cleaning the air on days when the spray water was cut off, a bank of filters was placed before the inlet to the air-conditioner as shown in Fig. 4, and the fan was operated to circulate the air through the filters, and to produce a circulation of the air within the room itself. Plots of effective temperature and relative humidity of the indoor and outdoor air are shown in Fig. 5. The

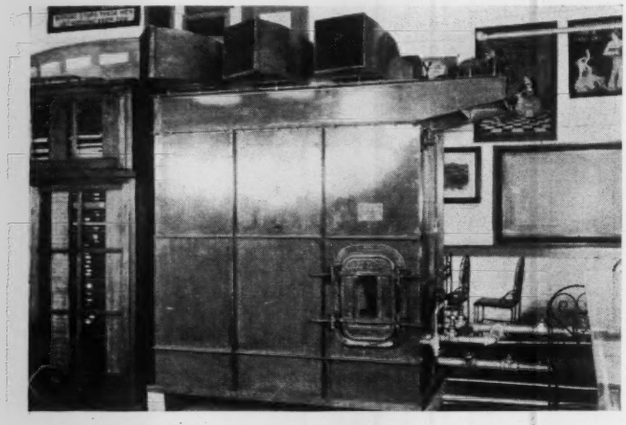

Fig. 3—Front View of Air Conditioning Unit

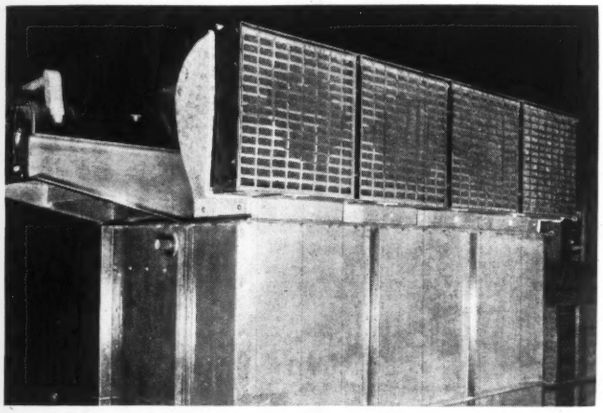

Fig. 4—Rear View of Air Conditioning Unit

values shown for each day are the average values for only the part of the day that the room was occupied, that is, for the period from 9:00 a.m. to 9:00 p.m.

The unit air-conditioner handled approximately 2000 cu ft of air per minute (at one of the lower fan speeds at which the unit was usually operated), creating about two and one-half air changes per hour by recirculation only. The air motion at the breathing line in Dicker Hall as determined by a Kata thermometer ran from 12 ft per minute to 41 ft per minute, averaging about 25 ft per minute. The effective temperatures shown in the table have been determined from the wet and dry bulb temperatures on the assumption of still air conditions only, the influence of air movement having been disregarded.

Before describing the result of observing the relief of hay fever symptoms in the air-conditioned room, some consideration should be given to the peculiarities of the disease called hay fever and to the experience which the medical profession has had with this disease.

As early as 1819 Bostock believed that hay fever was caused by "emanations from hay." As long ago as the middle of the 19th century Blackley noted that there were marked variations in the pollen content of the air during the hay fever season. He also noted that the severity of the symptoms in himself, as well as in other hay fever sufferers, depended chiefly on the concentration of the pollen in the air. This concentration he measured by counting the number of pollens on a unit area of an oiled glass slide exposed to the atmosphere for 24 hours.[4] Similar studies have been made by many since 1873 when Blackley published his classical paper Experimental Researches on the Causes and Nature of Catarrhus Aestivus.

Duke and Durham in 1928 wrote that, "The marked irregularity of the charts (pollen) often accounts for variation in the symptoms displayed by patients from day to day during a given season."[5] Rackemann and Smith of Boston[6] and Acquarone and Gay of Johns Hopkins have shown graphically and convincingly this correlation between severity of symptoms and the number of grains of pollen in the atmosphere to which the patient was exposed.[7]

An illuminating historical sketch is given by Thommer as follows:

When in 1819 Bostock published his now famous paper, Case of Periodical Affection of the Eye and Chest, he had already been a sufferer from the malady, later known as hay fever, for 38 years. Concerning his own treatment he tells us. "Topical bleeding, purging blisters, spare diet, bark and various other tonics, steel, opium, alterative courses of mercury, cold bathing, digitalis, and a number of topical applications to the eyes, have been fully tried." Though these measures were "persevered in with an unusual degree of steadiness" he informs us that "it is doubtful whether any distinct or permanent benefit has been derived from any of it ever."

It is apparent, therefore, that from the very outset, the disease must have been recognized as being inherently intractable. Bostock's estimate of the efficacy of the various therapeutic endeavors in his own case, is abundantly substantiated by the subsequent history of the treatment of the malady, for we find that during the next hundred years the manifold measures and multitudinous medicaments advocated as cures and as means of obtaining relief bear testimony by their very number to their general inefficiency and unsatisfactory character. The lack of adequate progress, not withstanding the numerous "certain cures" appearing in medical literature, was quite apparent.

In 1816 Henry Ward Beecher who had been a sufferer for many years wrote to Dr. Oliver Wendell Holmes as to his knowledge of a satisfactory remedy. Dr. Holmes could offer no encouragement.

After nearly a hundred years the quest for an efficient therapeutic agent was still being conducted unabated, for in 1903 Dunbar tells us that "I myself have tested all the hay fever remedies on which I could lay my hands within the last ten years but with no one of them did I accomplish a beneficial result."[8]

When certain pollens were recognized as the specific cause of pollen hay fever and pollen asthma, and specific skin tests for specific diagnosis were developed, specific treatment then became possible and promised to

supplant the "multitudinous medicaments," which at the best had only given temporary relief. With specific treatment cases have been greatly benefited and many have been entirely cured. There still remains, however, a large number of patients who, even in the hands of the foremost authorities on the treatment of these diseases have failed to receive any relief whatever from such specific therapy.

The *multitudinous medicaments* are still used by these unrelieved cases. In desperation they seize upon whatever straw of hope floats their way and no group of sufferers is more victimized by the vendors of patented medicines, and *guaranteed* cures which do not cure.

It is for this group that air conditioning offers the most satisfactory relief available. The size of the group can be appreciated from consideration of reports by leading investigators during the past decade. Pursuing a course of treatment aimed at desensitization, several investigators have secured varying degrees of satisfactory results as shown in the following table.[9]

Investigator	Percentage Results Consolidated		
	Perfect	Satisfactory	Poor
Cook and Vander Veer	12	52	36
Vander Veer	23	49	28
Walker	5	52	23
Rackemann	10	66	24

The results show that under the best of medical attention perfect results were obtained in only about one-fourth of the total number of cases treated. The remainder, or about three-fourths of all cases can be conceived of as having use for air conditioning at some time during their hay fever season.

The severity of symptoms varies greatly from one patient to another. The same disease that causes only an occasional more or less comical sneeze in one patient may make an invalid of the next, cause great economic loss through disability and expensive measures taken for relief, and if not relieved may cause premature death. Gay has recently pointed out the great economic loss from pollen disease disability and the public health importance of some kind of treatment for the child so affected in order to prevent the development of perennial asthma with secondary upper respiratory infection.[10] Such infections may result in permanent disability, and if unrelieved, threaten the life of the patient as in the case of Van Leuween's first case to be treated by air-conditioned atmosphere.[11]

Scheppegrell in 1924 described a clumsy apparatus for filtering pollen from air supplied to sufferers from pollen disease.[13] Van Leuween in 1925 reported success in having "miasma free" chambers built in the homes of patients who suffer with asthma. The method was so successful that it later became used as an official method of treatment in his clinic.[11] Van Leuween built hermetically sealed rooms, but other workers have found this precaution unnecessary if a slight positive pressure is maintained to prevent the infiltration of air borne pollen. The relief of bronchial asthma by more complete air conditioning was studied by Leopold and Leopold, whereas former reports dealt with simple filtration only.[15] Cohen,[16] Peshkin and Beck[17] treated hay fever by simple filters designed to remove pollen from the air and also reported success.

More recently Nelson[30] and his coworkers, and also Gay[18] have reported observations upon patients hospitalized and under controlled conditions allowing prolonged observation of patients under medical supervision. The former were successful in giving relief to some 83 per cent of uncomplicated cases of hay fever within 3½ hours.

In evaluating the benefit obtained by the patient it may be well to consider more than merely the temporary relief of symptoms. Gay[10] has pointed out the public health significance of the neglected case of pollen disease and the serious consequence which follows. It is of interest to note in this respect that when Van Leuween constructed his first "miasma free" chamber he was inspired not merely by the sufferings of a case of bronchial asthma, but by the attacks of bronchial pneumonia which all too constantly accompanied attacks of bronchial asthma in a child of 12 years. The encouraging results obtained by Van Leuween in this first case encouraged him to have other patients build such rooms in their homes to which filtered air could be supplied. Thus air conditioning may serve not only to relieve discomfort, but to preserve health in those patients who have not been relieved by other therapeutic measures and who are unable to seek climatic relief.

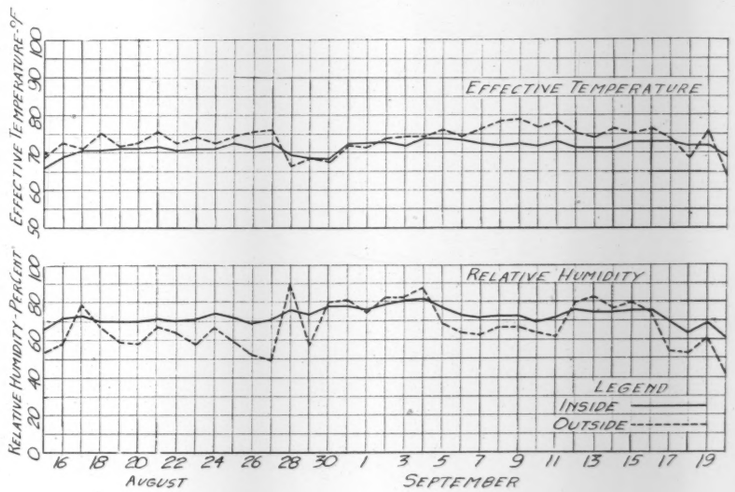

Fig. 5—Temperature and Humidity in Air Conditioned Room and Outside

The favorable reports of relief of hay fever and asthma by the use of conditioned air have caused air conditioning to be widely seized upon as a cureall, which, of course, it is not. It is a splendidly successful method of giving relief to certain patients who suffer from hay fever or asthma caused by air borne substances, particularly pollens. Enthusiastic proponents of air conditioning for hay fever and asthma relief make generous claims which may be difficult to substantiate in many cases. As an agent of relief for hay fever and asthma, its usefulness is limited, because all cases are not caused by air borne substances.

As early as 1830 Elliotson[19] described symptoms of a woman sensitive to rabbits and in 1864 Salter[20] described his own symptoms upon coming in contact with cats. Seasonal coryza and asthma due to emanations from sandflies has also been described,[21] as have hay fever symptoms due to cedar,[22] may flies,[23] orris root,[24] dog hair,[25] horse hair and horse dander. Even cuttlefish bones and bed bugs have been reported to have caused attacks of asthma.[24] Hay fever symptoms have also been reported as caused by the sting of a bee and by attacks of malaria, by animal hair used in furs and by many chemicals used in various industries. The need for specific diagnosis is apparent; and the enthusiast for air conditioning, or simple filtration, should not place himself in the position of guaranteeing relief. Such unfounded optimism can result only in frequent disappointment to the patient and discredit to the principle of air conditioning which can but delay its maximum usefulness as the valuable therapeutic adjunct which it is demonstrating itself to be.

The reports of others have indicated how many hay fever cases receive relief and how soon such relief is received under hospital conditions both as to the medically supervised selection of patients and their care and observation under treatment.[16-17-18] In the work reported upon in this paper the problem was to determine how much relief could be received by cases suffering with hay fever symptoms who were allowed to visit an air conditioned room at will. In studying the records an attempt has been made to determine the relative relief received by the visitors with respect to the different types of symptoms which the visitors experienced at the moment of entering the air conditioned room.

During the 1933 season approximately 100 hay fever cases visited Dicker Hall. Among these there were 40 cases selected for study. The remainder were excluded either because the records were not complete, or because the records showed that the cases were complicated by other disease. Some persons visited the hall once and left without experiencing relief, and there was no opportunity of following these cases for the completion of satisfactory records. The persons visiting the hall constituted a self-selected group, and came because they wished relief and returned only if they found it. Therefore, the records which have been considered as suitable for study are the records of those who experienced enough relief to make it worth their while to remain in Dicker Hall or to return on succeeding days, and for whom sufficient data were obtained.

The personal records consisted of a brief history of each patient and the type and degree of symptoms present upon entrance and at intervals during the patient's stay in the hall. The various symptoms observed and recorded were as follows:

Symptom Number	Part Affected	Type of Symptom
1	Eyes	Discomfort felt by patient. Itching, burning, sensitiveness to light
2	Eyes	Observed—lachrymation
3	Nose	Discomfort—inside; itching, burning
4	Nose	Congestion, blockage to breathing
5	Nose	Discharge
6	Nose	Discomfort outside; skin of nose, itching. Also other areas of skin as indicated
7	Sneezing	
8		Difficulty of respiration, asthma
9	Mouth	Discomfort, itching, burning
10	Throat	Discomfort, itching, burning
11	Ears, Eustacian tubes	Discomfort, itching, burning
12	Ears	Congestion causing interference with hearing
13	Headache	
14		General depression, fatigue, lassitude

In the discussion which follows, the symptoms have been designated by the corresponding numbers appearing in the first column of the preceding table. Symptoms 8, 11 and 12, however, have been omitted from further discussion and from the remaining tables because these symptoms occurred so seldom as to be insignificant. The subjective symptoms reported by the patient were rated according to severity as follows. Objective symptoms were rated in like proportion.

0 Feels as it does out of hay fever season.
1 Minimum, barely observable, not annoying, or if so because of duration rather than degree.
2 Definite and annoying.
3 Painful—Patient is unable to work effectively.
4 Patient is prostrate.

The next table shows the number of visitors, among the 40 selected for study, who complained of a given symptom at some time during the season, and also the percentage of the visitors complaining.

Symptom	Number of Visitors Complaining	Percentage of Visitors Complaining
1	29	72.5
2	19	47.5
3	25	62.4
4	36	90.0
5	32	80.0
6	11	27.5
7	27	67.5
9	6	15.0
10	10	25.0
13	9	22.5
14	21	52.5

This table shows that more visitors complained of nasal blockage (90 per cent) and nasal discharge (80 per cent) than of any other symptom. Next in frequency was irritation of the eyes, followed in order by sneezing, irritation of the mucous membrane of the nose, general depression and fatigue, watering of the eyes, itching of the skin, irritation of the throat, headache, and irritation inside the mouth.

During the 32 days on which records were made, the average number of days attendance in Dicker Hall amounted to only 3.1 days per visitor for those 40 visitors selected for study, making a total of 123 visitor-days, or an average of 3.8 visitors per day. The following table is a consolidation of the daily records and shows the number of patients, or visitors, who visited the air conditioned room on those days when the pollen content of the outside air was as indicated by the ar-

bitrarily-divided ranges of pollen counts as shown in the first column. The table shows also the number of symptoms complained of and the number of symptoms per patient.

Outdoor 24 Hour Pollen Count	Number of Days Upon Which Patients Visited Dicker Hall	Number of Symptoms Complained of	Number of Patients Visiting Hall	Number of Symptoms per Patient
0–24	7	49	14	3.5
25–49	7	132	32	4.1
50–99	10	221	53	4.2
100 up	8	105	24	4.4
Total	32	507	123	4.1

The number of symptoms increases from an average of 3.5 on days with pollen counts under 25 grains to 4.4 symptoms for days with pollen counts of 100 or more per 24 hours. The difference is not as marked as might be expected from the experience of others who have shown how symptoms increase and decrease with the variation in pollen count.[5,6,7] However, in these studies by others the measured severity of symptoms was not only for a given time, but also for a whole 24-hour period; whereas, in this study, the severity of a symptom refers to its severity at a given moment, that of entrance into the air conditioned atmosphere.

Durham[26] has shown how much the pollen concentration varies from hour to hour. Such variation makes it possible for an individual to have few symptoms when the pollen concentration is low and to be attacked a short time later the same day when the pollen concentration may have become much higher due to wind or other causes.

The 24-hour count is a measure of the time that the pollen concentration lasts as well as of how great a concentration occurs. Symptoms at any one time, however, depend mostly upon how much pollen is present at the moment. Pollen counts at the moment of occurrence of symptoms would be of interest, but difficult to ascertain by the ordinary method. The experience of having a severe attack following massive exposure to pollen, such as in an open field, is well known.

It may be, however, that, above a given threshold concentration of pollen, relatively low counts such as from 25 to 50 may be found to cause practically the same symptoms as higher counts up until very great concentrations are reached. The data given in the table would seem to support this. If this is true it becomes more important for air conditioning to reduce the pollen concentration as near zero as possible and not to be content with relatively low counts. Relatively low 24-hour counts may allow relief in hay fever resorts over a prolonged period, but for patients visiting an air conditioned room for short intervals the patient's judgment of relief is formed within a short time and the pollen concentration at the moment is important.

The accompanying table shows the frequency of the various symptoms based on the number of patient-visits. Also, the cases have been divided into the numbers which occurred under the second, third and fourth degree of severity. The first degree of severity has been ignored because the relative mildness of the symptoms made it difficult to note any marked change in relief.

Symptom	Degree of Severity			Total Frequency of Occurrence
	2	3	4	
1	61	16	1	78
2	50	3	1	54
3	48	14		62
4	68	43	5	116
5	73	33	1	107
6	14	3		17
7	40	27		67
9	11	1		12
10	17	5		22
13	10	1		11
14	35	8		43

This table shows, as before, that the most frequently occurring symptoms were nasal blockage and nasal discharge, and also that these symptoms were relatively more severe than other symptoms.

In attempting to estimate the relief caused by the air conditioning, certain assumptions must be made which cannot be absolutely substantiated. For example, it cannot be said with certainty that the patient's symptoms would have continued with undiminished severity if he had not sought the relief of conditioned air. Not uncommonly patients gave a history of having had more severe symptoms earlier in the day and that these symptoms had subsided spontaneously before coming to the hall.

Hay fever symptoms notoriously tend to occur spasmodically and attacks tend to terminate spontaneously without apparent cause. Therefore, it cannot be claimed that all of the relief experienced by patients while in the air conditioned atmosphere is caused by the air conditioning.

Also there were numerous circumstances which tended to reduce the degree of relief offered by the air conditioned atmosphere. Patients frequently would leave the hall to visit other portions of the building or even to go outside for short periods and thus delay their relief. When these excursions were for a matter of minutes only, no account was taken of them in the analysis. A most important factor was the possible presence of other unrecognized complicating diseases. Exposure to pollen within the room arose from the fact that it was impractical to prevent carrying pollen on clothing and also because there was some infiltration of unfiltered air. And yet, in spite of these circumstances, the capacity of the air conditioning unit to handle the room air volume several times an hour kept the pollen counts consistently low.

The possible introduction of pollen has been disregarded in this study because no means was provided for testing the theory that patients may have been affected by temporary increases in the amount of pollen in the room. Such increases could conceivably be sufficient to cause symptoms and yet not be sufficient to appreciably affect the pollen count on a 24-hour slide.

In any event, whatever factors tended to influence the symptoms of the patients, only the air conditioning and one other influenced them favorably. All other factors mentioned tend to retard relief. The one other influence was the use of drugs by patients immediately before and during their visit to the hall. Control was impossible; allowance difficult. In the tabulation the use of medicines has been disregarded. Unmedicated cases showed favorable results that tend to justify this policy.

A consolidation of the results obtained by observing

the relief secured by the visitors is given in the following table:

Symptom	Per Cent Relief	Time in Minutes Required for Relief
1	65	56
2	54	51
3	59	51
4	40	76
5	62	61
6	66	46
7	91	30
9	85	69
10	50	100
13	47	108
14	45	83

The foregoing table also shows the length of time patients were in the hall before maximum relief occurred. Symptoms receiving maximum relief under one hour were sneezing, itching of skin, watering of eyes. Other symptoms required over one hour. The time required for relief is important to the patient who has a limited time to spend in an air conditioned atmosphere.

As shown by Fig. 2, the pollen count in Dicker Hall varied on different days from 0 to 14 grains. An attempt was made to correlate the relief obtained in the hall with the inside pollen count. A summary of the results is given by the accompanying data:

Pollen Count Inside Hall	Visits Giving Some Degree of Relief	Visits Giving No Relief	Percentage of Visits Giving No Relief
1–4	377	35	8.49
5–9	117	14	10.68
10–14	38	7	15.55
Total	532	56	9.52

The data show that there was a consistent increase in unrelieved cases corresponding with the increase in the pollen counts. While not convincing these data strongly suggest that even small amounts of pollen interfere with relief in an air conditioned atmosphere. If this is true it makes it more desirable to reduce the amount of pollen introduced into the room on clothing and by infiltration.

Durham[27] has reported that pollen counts over a large area have chronological peaks corresponding to wind and storm in the area. In the present study it has been shown that rain causes an increase in the pollen count, especially when followed by sunshine. Observations over shorter periods than 24 hours are needed to determine the variations before, during and following rain. During this season a protracted rainy period was accompanied by low pollen counts. Counts seem to be increased by the wind before and the sunshine following, but decreased by the rain itself.

Moskow and Spain[28] have noted that the patient's discomfort from hay fever is decidedly increased upon days of very high temperature and low humidity. Their studies did not include estimates of pollen concentration.

In order to study the relationship between the relief of symptoms and the conditions within Dicker Hall an analysis was made of the relief obtained in relation to various conditions of temperature, relative humidity and pollen counts for symptom No. 4 which most frequently failed to be relieved. The analysis shows that failure of relief of the symptom occurred practically throughout the entire range of temperatures, relative humidities and pollen counts for which relief was experienced. If the data justify any provisional conclusion it is that effective temperatures below 70 deg are less favorable for the relief of nasal congestions than the higher temperatures. A summary of the relevant data is shown in Table 3.

Table 3

Number of Pollen Grains per Sq Cm per 24 Hr	Effective Temperature F	Degree of Relief Obtained							
		No Relief or Made Worse				Satisfactory Relief			
		% Relative Humidity				% Relative Humidity			
		65 to 69	70 to 74	75 to 79	80 to 84	65 to 69	70 to 74	75 to 79	80 to 84
1 to 4	65 to 69	1	0	1	0	1	4	2	0
	70 to 74	3	6	5	2	12	24	12	7
5 to 9	65 to 69	0	1	1	0	0	3	0	0
	70 to 74	0	0	0	0	6	10	6	0
10 to 14	65 to 69	0	0	1	1	0	0	2	1
	70 to 74	0	1	0	0	0	1	0	0

Considering the relative humidity alone, the results show that for a relative humidity of from 65 per cent to 69 per cent the percentage of visits giving relief to nasal blockage amounted to 88 per cent. The relief obtained became less as the humidity increased so that for humidities above 80 per cent the percentage of visits giving relief amounted to only 70 per cent.

If all symptoms were equally unpleasant it would be desirable to plan relief for those which occurred most frequently. Actually No. 4, nasal blockage, was one of the most distressing symptoms, occurred most frequently and was least often relieved.

Patients frequently reported that on the day following their visit to the hall that their symptoms were less severe than usual. Cause for such milder symptoms is difficult to determine. Symptoms have been demonstrated to vary widely from day to day, but to follow the rise and fall of the pollen count. Blackley[4] was able to predict from his own symptoms each day the approximate amount of pollen he would find on his slides. The Rev. Henry Ward Beecher noticed over 50 years ago that his symptoms seemed to be worse on alternate days. The variation in the daily pollen counts is the now accepted explanation of this irregularity of symptoms. Patients, however, experienced relief after a previous day's visit when the pollen count was too high to account for the relief. This suggests a prolonged benefit due to exposure to the air conditioned atmosphere, but there was no opportunity to study this aspect of the problem in detail.

Duration of relief could well be studied from the standpoint of the time of day the patient left the shelter of the hall and the outdoor pollen count at that time. Such pollen counts for special times were not available for this study. Durham[26] found that such counts made at hourly intervals through the day showed the count to be above the mean concentration from about 9 a. m. to 9 p. m. and below from 9 p. m. to 9 a. m. When patients visited on consecutive days and when data could be obtained with reasonable accuracy a record was made of which symptoms returned after the patient left the hall and what time lapsed before the return of such symptoms. It was observed that patients who left Dicker Hall in the evening did experience relief for a longer time than those

who left during the day. This is what one would expect from Durham's findings. It must be noted, however, that the patient who left in the evening was more likely to go indoors soon after leaving.

It was noted that when symptoms started shortly before the patient entered the air conditioned atmosphere or when patients left the comparative shelter of an indoor atmosphere to come to the hall, symptoms occasionally became worse for a short time after the patient entered. If the patient remained as long as one hour symptoms usually subsided but occasionally patients experienced increased symptoms and left the hall in discouragement without allowing sufficient time for a satisfactory test.

As the result of experiments of blowing dry pollen into one nostril of hay fever patients Efron and Penfound reported that most cases react within a few minutes after the test is performed, but symptoms in some cases were observed to be delayed as long as fourteen hours after this test.[29] Occasional increased discomfort experienced after entering the air conditioned atmosphere may be partially accounted for by a delayed action of the pollen received while exposed in the open air. Nasal blockage especially was the cause of increased discomfort not only to patients upon entering the hall, but also after prolonged visits within the hall. It was noted that this symptom tended to reoccur in the same individuals and it would therefore seem to be dependent upon some peculiarity of the individual patient. The fact that other symptoms could be relieved in these cases showed that the air conditioning was effective. The persistence and even aggravation of nasal blockage in occasional patients impressed the observers with the need for a thorough medical study of each patient and a conservative policy with respect to assuring a patient that air conditioning will relieve all of his symptoms.

Even with what seemed to be favorable cases, air conditioning may not relieve all symptoms because of some unsuspected complicating disease in addition to the hay fever. Occasionally patients were recognized to be suffering from some concurrent infection that interfered with their hay fever symptoms. Such conditions as head colds, nasal polyps, hordeolum, peri-tonsillar abscess, etc., cause the persistence of symptoms which otherwise would probably be relieved if due to uncomplicated hay fever. When such conditions were recognized the records were not used in this study. Without doubt many such complicating conditions were overlooked because it was not possible to obtain a complete medical diagnosis. Such undiagnosed cases included in this study tend to reduce the measure of relief obtained below that which uncomplicated cases might expect.

One such case is of special interest because it suggests the possible use of an air conditioned atmosphere as a diagnostic aid to the physician contemplating nasal surgery, and a protection to the patient following such surgery. A patient with hay fever presented himself with symptoms of nasal blockage. The physician wished to avoid surgery during the hay fever season for two reasons. First, because of the difficulty of determining how much of the disease was caused by a condition to be benefited by surgery and how much was caused by the patient's hay fever; and second, because nasal surgery during the hay fever season has been reported as a cause of hay fever and asthma. When the patient experienced relief from all hay fever symptoms except nasal blockage in an air conditioned atmosphere, the condition was considered a surgical one and the surgeon operated with good results. Following the nasal operation the patient spent as much time as possible in the pollen free atmosphere of the air conditioned room, a procedure which might well be made routine following nasal and throat surgery during the pollen season.

A second case is of interest because the patient suffered with little relief on the first and second day of her visit and was much discouraged with air conditioning until it was discovered on the third day that she was suffering from an abscess in her throat. After suitable treatment of the throat, her hay fever improved.

It has been noted that subjective symptoms are relieved in an air conditioned atmosphere more quickly than are objective symptoms.[30] This has been explained as due to the persistence of the tissue injury for a limited time after the cause of the injury (the pollen) has been removed. The same report called attention to the observation that at the onset of the hay fever season the offending pollen was found to be present in the air in appreciable amounts for days before patients complained of symptoms. Likewise the pollen disappeared from the atmosphere for a period as long as two weeks before patients were relieved of the discomfort caused by the prolonged pollen irritation of the mucous membranes. It has likewise been noted by patients that if they delay going to their hay fever resort until symptoms have started, that while they obtain early relief a certain amount of residual discomfort persists for a period of a week or more. One should therefore not expect air conditioning to immediately relieve these symptoms due to changes in tissue when these same symptoms are not immediately relieved by hay fever resorts nor by the absence of pollen at the end of the season.

Thus a number of patients failed to experience complete relief even after prolonged visits. If their relief were equal to that experienced during the days immediately following the close of the season it would seem logical to give greater credit to air conditioning than the simple record would indicate. These patients who are not entirely relieved may be those who have developed tissue changes which in turn cause secondary symptoms, and indeed from the history of a few cases this seems probable. Even though not complete this relief is often marked.

Another cause of only partial relief of symptoms may be that some cases have hay fever out of the pollen season, but find that symptoms are made worse by pollen. The removal of the pollen in such cases would be expected to give only partial relief and again the need for a medical history and diagnosis and a conservative policy is indicated. Here also, however, the relief obtained is definite and valuable.

Some cases seeking relief from symptoms of hay fever, are actually made worse by the air conditioned atmosphere. This observation has been noted before [18] and again impresses one with the fact that air conditioning should not be offered indiscriminately to all types of patients, suffering with what appears to be hay fever or asthma. Elderly persons and hay fever patients who also suffer with asthma may be made worse by the refrigerated atmosphere and for these cases simple air filtration may be advised.

Opportunity was offered occasionally for patients to compare the relief obtained in a private residence using a single air filter with the relief obtained in the air conditioned room. In one such case the patient experienced greater relief in the air conditioned room even though the pollen count made at the residence was lower than that in the cooler air conditioned room. The pollen counts were 1 and 0 for the residence and 2 and 3 for the air conditioned room. The temperature in the air conditioned room however was some degrees lower than that in the residence and this seemed to make the difference to the patient.

Two patients upon their first visit entered the hall while the spray water was turned off and the air was passing only through the filters. After 45 minutes they had experienced no relief and expressed disappointment at this time. Without notice to them the water spray was put into operation. Both patients remarked upon the decrease in their symptoms immediately after the spray water was started. They were near the machine and in position to notice the change in the temperature of the delivered air.

It was impossible to determine the comparative relief experienced by the same patient in the refrigerated atmosphere of the Hall and in a room adjacent to Dicker Hall equipped with a simple filter because patients rebelled against remaining in the warm room when the cooler room was available.

Efron and Penfound have reported that when dry pollen was introduced experimentally into the nostrils of patients most cases reacted within a few minutes.[29] Unfortunately for the patient seeking and finding relief in an air conditioned room away from his home, this experimental reaction has its counterpart in nature and symptoms frequently return with distressing promptness after the patient leaves the air conditional atmosphere. Fränkkel[31] has described a portable filter mask which might prove useful to patients going out of doors for short periods.

It was noted that the time required for the return of symptoms varied not only from patient to patient on the same day, but also for the same patient. Other than the patient's tolerance, the pollen concentration was probably the most specific factor involved.

Many hay fever sufferers have noted a comparative mildness of symptoms as long as they remained within their homes. They have noticed that by reducing the ventilation of their homes they can reduce the severity and the number of their symptoms. If such patients have severe attacks upon going out of doors they plan to remain in doors as much as possible. This may be the very type of case that is most relieved by a visit to the air conditioned room, but because of the attack suffered while on the way back home the patient prefers to remain at home for days at a time rather than to risk the more severe attacks which follow the greater but temporary relief experienced in the air-conditioned room.

Summary

The results of this work may be summarized by the following statements which appear to be substantiated by the personal records of the visitors to Dicker Hall:

(1) All of the 40 visitors selected for study received some degree of relief in the air conditioned room. Satisfactory relief was obtained upon about 90 per cent of all visits made by these patients.

(2) Records of other visitors were unsuitable for study because in some cases symptoms were of a degree too mild for the observation of relief, and because in other cases in which relief was obtained the patient either could not be kept under observation or other pertinent data could not be obtained.

(3) Some cases received relief while in the room but failed to visit the room often enough to allow for sufficient study. Two principal reasons given for failure to visit the room were:

(a) lived too far away, and hence not convenient to come; and
(b) attacks appeared on the way to and from the hall, and hence preferred to remain at home.

(4) The various common symptoms usually experienced by hay-fever sufferers received different degrees of relief.

(a) Sneezing and irritation of mucous membrane received the greatest amount of relief.
(b) Irritation of the eyes, facial irritation and nasal discharge received a moderate amount of relief.
(c) Nasal blockage, headache, and general depression or fatigue received the least amount of relief.

(5) The data do not show conclusively how much relief is due to filtration only and how much is due to the more comfortable temperature and humidity which prevailed within the air conditioned room as compared with less comfortable conditions in other rooms. The fact that the patients *preferred* the air conditioned room would merely confirm the judgment of normal persons.

(6) There was occasionally an exceptional case that was made more uncomfortable in the cooler atmosphere of the air conditioned room.

(7) The observers were impressed with the need in all cases of medical diagnosis and sensitivity tests to determine the specific cause of symptoms.

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Comfort Cooling with Attic Ventilating Fans

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EXPERIMENTS in residential cooling, initiated in the summer of 1932, were continued during this past summer, and operating data were gathered on three residential cooling and ventilating installations which were directly sponsored by the Detroit Edison Co. This paper summarizes the season's operating experience with these installations and describes certain features of design and construction which, it is believed, will be of special interest at this time.

Cooling By Attic Ventilating Fans

The costs of cooling installations which have sufficient capacity to cool artificially a moderate-sized residence are still high enough to restrict their use to a comparatively few people, while the desire for more comfortable conditions in the home during the hot summer season is almost universal. If a simple and relatively inexpensive means can be made commercially available, whereby at least the sleeping rooms may be made more comfortable at night, there unquestionably would be a broad market for such equipment.

This suggests the development of a method by which the outdoor air can be drawn through the rooms at such times as the temperatures are lower than those indoors. A study of the summer temperature records for Detroit shows that, almost invariably, the outdoor temperature drops below the inside temperature at around 7 to 8 p.m., and stays below for approximately 12 hours. The attic ventilating fan seemed to give such promise as a means of accomplishing this purpose that three types of residential installations were made during the past year and a half and considerable experience has been gained with their operation. The attic ventilating fan is by no means a new idea, but its possibilities seem to have been quite overlooked by nearly everyone.

In addition to providing more comfortable conditions for the second floor of the home, it was felt that an attic fan could be used to pull air through the first floor rooms during the early evening and thereby accomplish an appreciable improvement to the inside air condition usually existing after a hot summer day. It is then just one step further to install an attic fan of sufficient capacity to pull a large volume of air through both the first and second floor rooms and so cool down the building structure during the night that, with no artificial cooling during the following day, the conditions will be tolerable. It was with the thought that the attic ventilating fan might serve as a substitute for artificial cooling in many residences, where a moderate improvement to comfort would be most welcome, that this experimental work was undertaken.

The Fan Installations

The installations made in the Detroit residences were of three distinct types. One was a standard commercial propeller fan, the second a centrifugal multi-blade fan, and the third a centrifugal, steel plate, curved blade fan, installed without a scroll. The essential requirements for cooling systems of this kind are quietness of operation and low first cost. Fan efficiency is unimportant because of the comparatively short season of use.

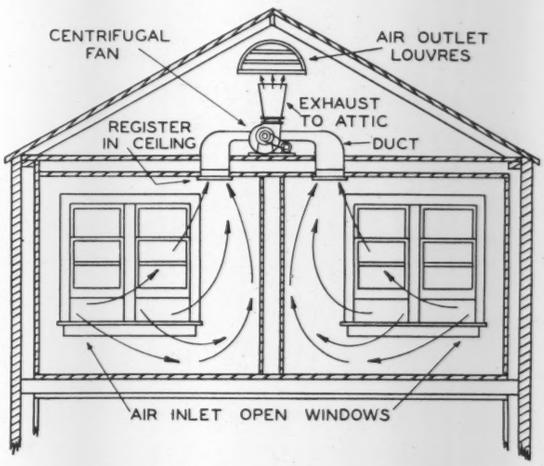

Fig. 1—Centrifugal multi-blade fan installed with exhaust ducts

Centrifugal Fan: A multi-blade centrifugal fan was installed in the attic space of an eight room house of brick and stucco construction. The suction side of the fan was connected to exhaust ducts leading to grilles placed in the ceilings of two bedrooms, the air being drawn through the open bedroom windows and discharged into the attic space where it flowed to the outdoors through louvres placed in the gables. The general arrangement is shown in Fig. 1 and the principal data for the installation are shown in Table 1.

Propeller Fan: A 10-blade propeller fan of the disc type was installed at the head of the attic stairway of a six room house of ordinary frame construction. The fan was mounted over the attic stair well in such a manner that the entire attic stairway is placed under suction. The air is drawn through the open windows of the three bedrooms into a central hallway, and thence up the attic

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Table 1—Comparison of Attic Ventilating Fans in Residences

Type of Fan	Centrifugal Multi-Blade (Exhaust Duct System)	Propeller	Centrifugal Steel Plate Curved Blade—No Scroll
Diameter of Wheel—Inches	18	21¼	36
Capacity—cfm (as measured)	2754	3060	5040
Fan Speed—rpm	300	900 600	320
Blade Tip Speed—fpm	1415	5000 3300	3010
Motor Drive	½ hp Capacitor Start and Run—1140 rpm—V-Belt Drive	Direct Connected to a 1/6 hp 900 rpm A-C Motor	1/3 hp Capacitor Start and Induction Run—1715 rpm—V-Belt Drive
Static Pressure of Fan—Inches of Water	0.10*	0.038**	0.11*
Volume of Space Ventilated—cu ft	4600	5000	5800
Rate of Air Renewal—Changes per hour	36	37	52
Fan and Motor Mounting	Sponge Rubber Insulated Wood Platform	Bolted to Wood 2x4's faced with strips of felt	Welded Angle Iron Frame supported at four corners on 2″x2″x1″ sponge rubber pads
Power Input—Watts	320	252	375
Cfm per Watt Input	8.6	12.1	13.4
Power Consumption for Season 1933—kw-hr	48	14	....
Hours of Operation—1933	150	55	....

*Static pressure of fan is computed to be the static pressure at fan outlet plus static pressure (draft) at fan inlet minus velocity pressure at fan inlet; draft gauge mounted in the attic in all cases.
**Static pressure readings, on the suction and discharge side of the fan, were taken where air velocities were negligible, consequently no correction for velocity head is made; draft gauge mounted in the attic.

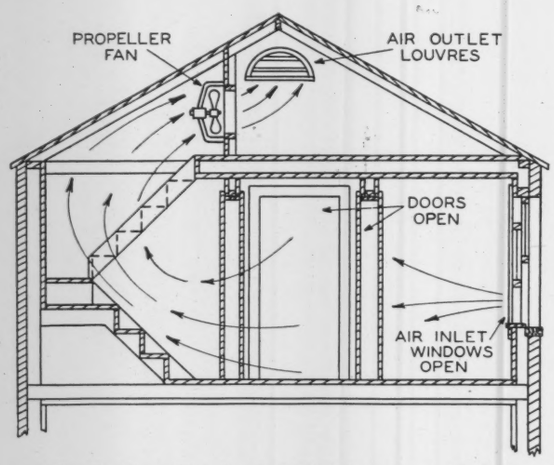


Fig. 2—Propeller fan installed in attic stair well

stairs where it is discharged into the attic space and allowed to find its way to the outdoors through open windows placed at either end of the attic. The general arrangement is shown in Fig. 2 and the installation data are given in Table 1.

Centrifugal Steel Plate Fan: In an endeavor to avoid some of the noise experienced with propeller fans, and at the same time overcome the obstacle of higher installation costs which are inherent in a system requiring duct work connected to a centrifugal fan, a rather unique fan assembly was devised. This assembly, shown in Fig. 3, consists of a single inlet, backward curved blade, steel plate type, centrifugal fan wheel, mounted on a light angle iron frame which supports the fan wheel, shaft, two bearings, V belt drive, and motor. No fan scroll was used and the inlet was placed close to a circular opening cut in a plaster board partition which served to divide a small attic space into a suction and discharge chamber. As in the case of the propeller fan installation, air is drawn up the attic stairway, into the fan wheel, and discharged freely from the blades into a small attic space, from whence it flows through open attic windows to the outdoors. The installation data for this fan are also to be found in Table 1.

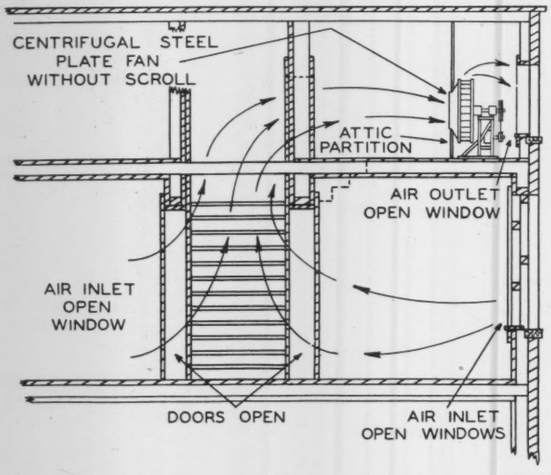


Fig. 3—Centrifugal curved blade steel plate fan installed without scroll

Operating Results

The operating experience gained with these three installations during this past summer has shed considerable light on the possibility of popularizing this type of comfort cooling, and the data and experience will be discussed in the order in which the installations have just been described.

Centrifugal Attic Ventilating Fan: The centrifugal fan illustrated in Fig. 1 was operated about 150 hours during the past summer, and the occupants of the house were well pleased with the results obtained. The temperatures in the two bed rooms to which the fan is connected were very responsive to the action of the fan, and the characteristic temperature curve for the larger of the two rooms is shown in Fig. 4. It will be seen that the inside temperature can be made to follow the declining outdoor temperature curve very closely, the temperature at midnight being only about 2 deg above that for the outdoors. This is in contrast to the conditions obtaining when the fan was not used, as the curves show the indoor temperature to be at least 8 deg higher than the outdoor at midnight, and additional temperature records show this difference to have been as high as twelve degrees on some occasions.

Anemometer readings taken at the ceiling grille showed an air velocity of about 330 fpm, corresponding to 1294 cfm passing through this room. As the volume of the bedroom was 2180 cu ft, this rate of air flow

provided about 36 changes of air per hour. With a total air flow of 2754 cfm through the fan, the static pressure of the fan was 0.10 in. water. The efficiency of cooling[1] at this rate of air change is shown quite effectively in Fig. 5. The indoor and outdoor temperature drop has been plotted at one hour intervals between the point at which the two temperature curves cross and the point at which the drop is a maximum; the latter being the point at which the actual temperatures reach

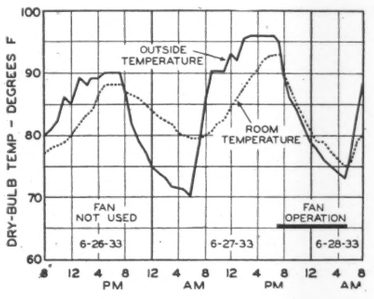


Fig. 4—Effect of centrifugal attic fan ventilation upon the temperature of an upstairs room

their minimum value. These data were taken from Fig. 4 for the evening of June 27. It is evident that sufficient outdoor air was being drawn through the bed room to make the indoor temperature drop quite uniformly with the outdoor, with a differential between the two temperatures of about two degrees. On the previous night, the night of June 26, the fan was not operated, and natural ventilation was depended upon to reduce the inside temperature. The curve for natural ventilation, shown in Fig. 5, indicates a very slow drop in indoor temperature in the early evening hours, when interpreted in terms of the actual temperature curves for June 26 as they appear in Fig. 4. Since the actual maximum drop in indoor temperature was less than half as great as that for the outdoor, and also less than half as much as the drop in inside temperature on the succeeding night when the fan was used, it seems fair to conclude that, in this installation at least, natural ventilation is only about half as effective in reducing inside temperatures as is the case with fan ventilation. This contrast is even more striking if consideration is given to the fact that, on the night the fan was used, both the day and night outdoor temperatures were considerably higher than obtained on the previous night when natural ventilation was employed. It should be pointed out, however, that in this installation, there was no opening into the attic, except through the ceiling grille, exhaust duct, and fan, so that natural ventilation was not aided by the stack effect of the structure, whatever real value that may have. The temperature records alone do not indicate the entire cooling effect produced. There was also a very noticeable air movement set up which decidedly augmented the temperature reduction.

Another illustration of the effectiveness of this fan is the comparison of bed room temperatures affected by the fan with the temperatures of another bed room not connected to the fan. This comparison is shown in Fig. 6. The unventilated bed room has a northeast exposure while the ventilated bed room has a northwest exposure. It will be seen that the temperature curve for the ventilated bed room drops below the curve for the unventilated room, even though in both cases it starts from a much higher maximum.

Although the occupants of the room reported that the diffusion of the air drawn through it was quite satisfactory, it was decided to check this diffusion by moving the thermometer to various parts of the room on succeeding nights. The thermometer was located in the position shown in Fig. 7 when the temperatures were recorded which are shown in Fig. 6. When the temperatures shown in Fig. 4 were recorded, the thermometer was located near the center of the room. Other locations were tried from time to time, and, except for the location near the ceiling grille, the recorded inside temperature drop showed very nearly the same relationship to the outdoor temperature drop in each case. This indicates a much better diffusion than was at first thought possible.

Propeller Fan: The propeller fan shown in Fig. 2 has a capacity, as determined by an anemometer, of about 3000 cfm when pulling air through the three bed rooms. This is equivalent to about 37 changes of air per hour. Under these conditions the static pressure of the fan is 0.038 in. of water. As the fan was operated over shorter periods than was the case with the centrifugal fan, the bed room temperatures did not fall nearly as far, but during the rather short periods of

[1]Calculated as suggested by Prof. A. P. Kratz and S. Konzo, Engrg. Exp. Station, University of Illinois.

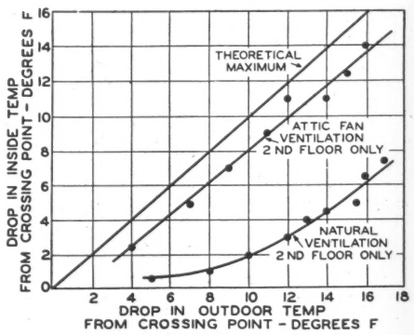


Fig. 5—Relation between inside and outside temperature drop following the point at which the temperatures coincide

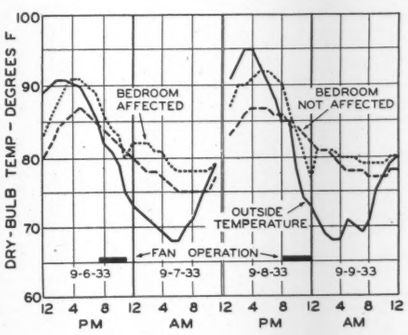


Fig. 6—Comparative temperatures in two similar bed rooms, one affected and the other not affected by attic ventilation

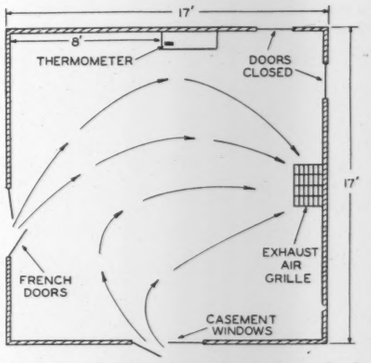


Fig. 7—Distribution of air in a bed room ventilated by an attic fan

operation, usually from about 8 to 11 p.m., the cooling effect was just as pronounced, as is evidenced by the curves shown in Fig. 8. It will be noted that, during the fore part of the evening of September 6, the fan was used to pull air through both the downstairs and upstairs rooms. When operating in this manner air was pulled through the downstairs rooms and up the stairway at the rate of about 10 air changes per hour, while at the same

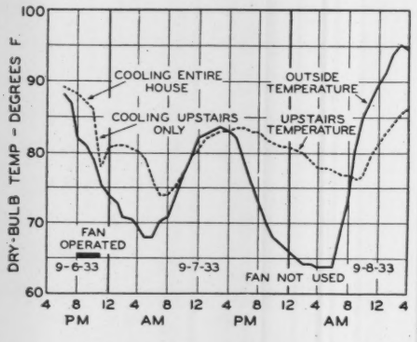


Fig. 8—Effect of propeller attic fan ventilation upon upstairs temperatures

time air was drawn through the bed rooms at the rate of about 20 changes of air per hour. The break in the upstairs indoor temperature curve indicates the point at which the downstairs windows and doors were closed, and from that time on the air was drawn through the bed rooms only. The rather sharp rise in indoor temperature, which takes place after the fan is shut down, is characteristic of these installations, and is undoubtedly due to the heating effect of the walls and ceiling. The inside temperature drop obtained when natural ventilation was used seems to bear about the same relationship to the temperature drop produced by the propeller fan as that obtained with the centrifugal fan previously described. When the fan was not used, the attic door was kept open as before, consequently the stack effect of the structure could be utilized. Although there was a noticeable natural draft up the attic stairway, the cooling effect produced was in no way comparable to the results obtained when the fan was operated.

On several occasions the fan was used during the early evening to ventilate the downstairs, and quite satisfactory results were obtained. To obtain best results, of course, the bed room doors had to be kept closed. The fan was especially effective in removing tobacco smoke from the living room when a group of people were present. If the bed room doors were closed, tests showed that it was possible to pull sufficient air through the downstairs rooms to produce about 30 changes of air per hour. When the downstairs was closed and only the bed room windows were open, there was not enough air movement up the stairway to the second floor to register on the anemometer. This does not prove, of course, that there was no leakage of air into the lower part of the structure, as the anemometer readings had to be taken at a point where the area was rather large. None of the tests, however, indicated that the leakage of air into the structure was serious enough to have an appreciable effect on the cooling and ventilating capacity of the fan.

Centrifugal Steel Plate Fan: The fan assembly shown in Fig. 3 was not installed until the latter part of last summer, consequently reliable temperature data are not available. Capacity tests were made, however, and enough information was gathered to give assurance that the fan was capable of effectively ventilating the three upstairs bed rooms to which the center hall and attic stairs are connected.

On the first test the fan was operated at 390 rpm, and anemometer measurements showed an air movement up the attic stairway of about 5880 cfm. At this speed, however, there was considerable air noise, and for the second test the fan speed was reduced to 320 rpm. At the latter speed the fan handled 5040 cfm up the attic stairway at a static head of 0.11 in. of water. With the downstairs closed up, and the windows open in two bed rooms, the air flow through one of the bed rooms was 1580 cfm and through the other 2600 cfm. With the volume of the second floor equal to about 5800 cu ft this rate of air flow provided about 52 changes of air per hour. The measured air flow through the two bed rooms which were open was 4180 cfm, and the difference between this volume and the quantity of air actually handled by the fan can be accounted for by in-leakage of air through windows in other rooms and leakage up the first floor stairway. The air movement up the stairway, however, was not sufficient to register on the anemometer. This leakage and short circuiting of air amounted to about 860 cfm, or about 17 per cent of the total volume of air handled by the fan. With this rather high rate of air change it seems certain, considering the experience with the other installations, that this fan will accomplish very effective cooling.

Control of Fan Noise

Since the attic ventilating fan is usually located directly over the sleeping rooms, it is quite essential that every possible precaution be taken to reduce the fan and motor operating noises to a level which is not objectionable to the occupants of the home. In the case of the centrifugal fan installation, both fan and motor were mounted on a wood platform which was insulated from the attic floor joists by strips of ½ in. sponge rubber, and rubber washers were used under all holding-down bolts. In spite of these precautions, the initial fan speed of 400 rpm proved to be too high for quiet operation, and it was reduced to about 300 rpm. It was also found desirable to limit the velocity of the air through the ceiling grilles to about 400 fpm in order to reduce the air noise. Motor noises have frequently proved troublesome, and the capacitor type of motor has been found particularly well adapted to this service because of its inherent quietness of operation.

The propeller fan installation described in this paper was not satisfactory from the standpoint of noise. At the normal operating speed of 900 rpm the blade tip speed was 5000 fpm and the air noise was quite disturbing. For that reason the fan usually was not operated after the occupants had retired for the night. Although strips of felt were placed between the fan casing and the wood studding to which it was bolted, this was not very effective in reducing the transmission of sound through the building structure. Experience thus far indicates

that the tip speed of a ventilating fan should not exceed 3200 fpm if quiet operation is desired.

Conclusions

1. It is quite practicable, by the use of either a centrifugal or propeller type attic ventilating fan, to very effectively cool the sleeping rooms in a moderate-sized house.

2. The downstairs portion of the house can be effectively ventilated if the fan is connected to an attic stairway in such a manner that the entire first floor can be placed under suction by way of the first floor stairs. It is not advisable, however, to leave the downstairs open after the occupants of the house have retired, as the flow of air through the bed rooms, and consequently the cooling effect in these rooms, is greatly reduced.

3. Wherever possible, the capacity of the fan should be such as to provide a minimum of about 30 air changes per hour. In very large homes, physical or commercial limitations may dictate the use of a proportionately smaller fan, providing about 20 changes of air per hour. This is still sufficiently high to accomplish reasonably satisfactory cooling.

4. Natural ventilation, induced by the stack effect of the structure resulting from the opening of attic doors and windows, is not nearly as effective in circulating air from out of doors as is an attic fan which provides about 30 changes of air per hour.

5. It seems reasonable to conclude, in the light of the past summer's experience, that attic fan ventilation for residences should prove to be a satisfactory substitute for artificial cooling during a large part of the summer season under climatic conditions similar to those in Detroit.

Report of the Committee on Research

By **G. L. Larson* and F. C. Houghten†**

THE Committee on Research of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS held two meetings in 1933, one last January in Cincinnati, and one in June in Detroit. Important items of business, including readjustment of the budget, were cared for by the Chairman of the Committee on Research, the Chairman of the Research Finance Committee, the Technical Adviser to the Committee, and the Director of the Research Laboratory at a meeting in Chicago in September. Additional business of the Committee was handled by correspondence.

A. S. H. V. E. RESEARCH LABORATORY, PITTSBURGH, PA.
BUDGET APPROPRIATIONS AND DISBURSEMENTS
For the Year Ended December 31st, 1933

REVENUES	Budget	Actual Receipts
Share of Dues from A. S. H. V. E.	$ 9,396.00	$ 6,203.02
Share of Dues from A. S. H. V. E. 1932	637.76	
Contributions, 1933 Pledges	5,000.00	3,524.00
Interest on Security Owned	135.00	135.00
Interest on Bank Balances	.....	8.25
Refund—American Society for Testing Material Exhibit	.....	108.14
Contributions—1932 Pledges	1,250.00	630.00
Interest on Endowment	1,000.00	.....
Cash from 1932	4,446.38	.....
Gross Revenues	$21,865.14	$10,608.41

EXPENSES	Budget	Actual Expenditures
Travel—Committee and Laboratory Personnel	$ 800.00	$ 401.92
Salary—Clerical Society Headquarters	300.00	.....
Salary—Technical Adviser	1,400.00	500.00
General Printing and Promotion	100.00	70.30
Correlating Thermal Research	500.00	500.00
	$ 3,100.00	$ 1,472.22
Laboratory at Pittsburgh—		
Salaries	$ 9,000.00	$ 8,491.70
Laboratory Supplies and Equipment	600.00	472.54
Office Supplies and Expense	400.00	344.63
	$10,000.00	$ 9,308.87
Contracts with Cooperative Institutions—		
University of Illinois	$ 4,150.00	$ 3,225.00
University of Minnesota	875.00	875.00
Harvard University	1,704.14	1,153.02
Yale University	750.00	200.00
Miscellaneous	.....	50.00
	$ 7,479.14	$ 5,503.02
Contingency Fund—Expenses		
Exposition Exhibit Expense	$ 1,286.00	$ 259.81
	$21,865.14	$16,543.92

*Chairman, A. S. H. V. E. Committee on Research, Chairman, Mechanical Engineering Department, and Professor of Steam and Gas Engineering, University of Wisconsin.

†Director, A. S. H. V. E. Research Laboratory.

For presentation at the 40th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, New York, N. Y., February, 1934.

Financial difficulties attending the continuation of the depression made it necessary to further curtail all activities in order to keep expenditures within funds available.

1933 Research Contributors

National Warm Air Heating and Air Conditioning Association
Utilities Research Commission, Inc.
Aerofin Corp.
American Air Filter Co., Inc.
American Blower Corp.
American Radiator Co.
Barber-Colman Co.
Barnes & Jones
Brillo Manufacturing Co., Inc.
Buffalo Forge Co.
Bush Manufacturing Co.
Carrier Engineering Corp.
Detroit Edison Co.
Detroit Stoker Co.
Donohoe Building Equipment Co.
C. A. Dunham Co.
Frigidaire Corp.
General Electric Co.
Hoffman Specialty Co., Inc.
Hough Shade Corp.
Illinois Engineering Co.
Independent Air Filter Co.
Johns-Manville Co.
Johnson Service Co.
Lakeside Co.
Marr-Galbreath Machinery Co.
Masonite Corp.
Moorhead-Reitmeyer Co., Inc.
Herman Nelson Corp.
J. J. Nesbitt, Inc.
New York Steam Corp.
Owens-Illinois Glass Co.
Pittsburgh Plate Glass Co.
Richmond Radiator Co.
Staynew Filter Corp.
B. F. Sturtevant Co.
Trane Co.
Union Electric Light & Power Co. of St. Louis
Waterman-Waterbury Co.
Westinghouse Electric & Manufacturing Co.
J. G. Wilson Corp.
Wood Conversion Co.
York Ice Machinery Corp.

Research During 1933

Sixteen Technical Advisory Committees were appointed by the Committee on Research to give consideration to the need for research and plans for the same in connection with as many technical subjects. These committees served both through meetings at the time of the Annual and Semi-Annual Meetings of the Society and through correspondence. Some of their reports have resulted in standardization and correlation of technical information in regard to the particular subject in which they were interested. Through their efforts programs for research were completely outlined in a number of instances and in other cases material progress was made. Because of lack of funds, active investigation of only a few of the subjects could be carried on at the Laboratory in Pittsburgh or in the co-operating universities. Prof. A. C. Willard continued his service as Technical Adviser to the Committee on Research.

Co-operation With the Universities

The Committee on Research continued the practice of enlarging its useful work through co-operation with the engineering departments of a number of universities. While it has been necessary to reduce and in many cases eliminate entirely the expenditure of funds for the support of work carried on in the universities, active co-operation has been continued with many of the institutions formerly co-operating, and co-operation has been organized in other institutions during the year. The following institutions co-operated with the Laboratory during 1933 in connection with the projects listed:

1. *UNIVERSITY OF WISCONSIN:* Heating of buildings.
2. *ARMOUR INSTITUTE OF TECHNOLOGY:* Study of air flow through registers and grilles and air delivery by unit ventilators.
3. *UNIVERSITY OF ILLINOIS:* Direct and indirect radiation with gravity air circulation. Also, cooling of buildings in summer.
4. *UNIVERSITY OF KANSAS:* Garage ventilation.
5. *HARVARD SCHOOL OF PUBLIC HEALTH:* Atmospheric ionization. Also, minimum air requirements for ventilation.
6. *YALE UNIVERSITY:* Oil burning equipment used in heating.
7. *UNIVERSITY OF MINNESOTA:* Dust and dust control apparatus. Also, heat transmission.
8. *AGRICULTURAL AND MECHANICAL COLLEGE OF TEXAS:* Insulation of buildings.
9. *CASE SCHOOL OF APPLIED SCIENCE:* Heat transfer from direct and extended surfaces with forced air circulation.
10. *MICHIGAN COLLEGE OF MINING AND TECHNOLOGY:* Corrosion in return lines in relation to the chemical composition of the water, vapor, and gas handled.

Papers Published During 1933

The Committee on Research through its various Technical Advisory Committees published 20 technical reports during 1933 in the monthly Journal of the Society:

1. Cold Walls and Their Relation to the Feeling of Warmth, by F. C. Houghten and Paul McDermott. January 1933.
2. Study of Summer Cooling in the Research Residence at the University of Illinois, by A. P. Kratz and S. Konzo. February 1933.
3. Summer Cooling Operating Results in a Detroit Residence, by J. H. Walker and G. B. Helmrich. February 1933.
4. Condensate and Air Return in Steam Heating Systems, by F. C. Houghten and J. L. Blackshaw. March 1933.
5. The Heat Conductivity of Wood at Climatic Temperature Differences, by F. B. Rowley. June 1933.
6. Indices of Air Change and Air Distribution, by F. C. Houghten and J. L. Blackshaw. June 1933.
7. Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work. Presented at the Semi-Annual Meeting of the Society in June 1933.
8. Application of the Eupatheoscope for Measuring the Performance of Direct Radiation and Convectors in Terms of Equivalent Temperature, by A. C. Willard, A. P. Kratz, and M. K. Fahnestock. July 1933.
9. Carbon Monoxide Distribution in Relation to the Heating and Ventilation of a One-Floor Garage, by F. C. Houghten and Paul McDermott. July 1933.
10. Physiologic Changes During Exposure to Ionized Air, by C. P. Yaglou, A. D. Brandt, and L. C. Benjamin. August 1933.
11. Tests of Convectors in a Warm Wall Testing Booth, by A. P. Kratz, M. K. Fahnestock, and E. L. Broderick. August 1933.
12. Temperature Gradient Observations in a Large Heated Space, by G. L. Larson, D. W. Nelson, and O. C. Cromer. September 1933.
13. Measurement of the Flow of Air Through Registers and Grilles, by L. E. Davies. September 1933.
14. Radiation of Energy Through Glass, by J. L. Blackshaw and F. C. Houghten. October 1933.
15. Comparison of Oil and Gas Firing in a Heating Boiler, by L. E. Seeley and E. J. Tavanlar. October 1933.
16. Study of Summer Cooling in the Research Residence for the Summer of 1933, by A. P. Kratz and S. Konzo. December 1933.
17. Carbon Monoxide Surveys of Two Garages, by A. H. Sluss, E. K. Campbell, and Louis M. Farber. December 1933.
18. 1933 Directory for the Correlation of Thermal Research. Published by the Research Laboratory-A. S. H. V. E., in booklet form under the direction of the Technical Advisory Committee on Correlating Thermal Research.
19. Diurnal and Seasonal Variations in the Small-Ion Content of Outdoor and Indoor Air, by C. P. Yaglou and L. C. Benjamin. January 1934.
20. Studies of Solar Radiation Through Bare and Shaded Windows, by F. C. Houghten, Carl Gutberlet, and J. L. Blackshaw. February 1934.

Research Projects Investigated During 1933

The membership of the sixteen Technical Advisory Committees and their activities during 1933 are given below:

1. AIR CONDITIONS AND THEIR RELATION TO LIVING COMFORT

C. P. Yaglou, *Chairman*
J. J. Aeberly
W. L. Fleisher
D. E. French
R. R. Sayers
C.-E. A. Winslow

Activities by the committee were continued on problems relating to atmospheric and artificial ionization, and minimum air requirements for ventilation. A progress report[10] on the physiological changes during exposure to ionized air was presented at the Semi-Annual Meeting, 1933, giving the results of experiments at Harvard University in co-operation with the Laboratory. The work is being continued independently by Harvard on patients of its clinic. A second report[19] dealing with diurnal and seasonal changes in atmospheric ionization both outdoors and indoors is being presented at the Annual Meeting, 1934.

A study of minimum air requirements for ventilation from the standpoint of body odor is now being made by Harvard in co-operation with the Laboratory. This is an elaboration of the ventilation-box experiments recently made at Harvard. Two identical rooms having a small connecting door in the adjoining wall are equipped with individual unit air conditioners for maintaining the same temperature and humidity in both rooms. The control room is to be furnished with 30 or more cubic feet of air per person per minute and will be occupied by 4 to 6 trained "air smellers" who will serve as judges. The other room will be occupied by 8 to 10 subjects, receiving a varying supply of air between the limits of 5 and 30 cu ft per minute per person. After a preliminary exposure of one to two hours or more, the judges will pass back and forth from one room to the other, making comparisons between the odor intensities in the two rooms. Notes will be made of any unusual or objectionable features that may be associated with unusually low or high air flows. As a check on the foregoing readings, observations will be made of the number of dilutions necessary to reduce the odor intensity to the olfactory threshold, by means of a simple device which has been developed in connection with the ventilation-box experiments.

2. AIR FLOW THROUGH REGISTERS AND GRILLES

John Howatt, *Chairman*
J. J. Aeberly
L. E. Davies
D. E. French
J. J. Haines

The work carried on by this committee was originally suggested to the Committee on Research by the *Ventilating Contractors Employers Association* of Chicago, which felt the need of a practical method of determining the rate of air delivery in an installed job. The cost of the investigation was largely borne by the contractors association, and the work was carried on by Armour Institute of Technology in co-operation with this organization and the Society. The work resulted in a paper[13] presented at the Semi-Annual Meeting, 1933. The investigation has lead to the development of formulae for air flow through both supply and exhaust grilles and a table of coefficients for use therein, which have become generally recognized and adopted as the best available practice. The publication of this year's work of the committee completes the studies originally outlined.

3. ATMOSPHERIC DUST AND AIR CLEANING DEVICES (INCLUDING DUST AND SMOKE)

H. C. Murphy, *Chairman*
J. J. Bloomfield
Albert Buenger
Philip Drinker
Leonard Greenburg
E. V. Hill
S. R. Lewis
H. B. Meller
Games Slayter
S. W. Wynne

Active work was carried on by this committee during the year, both in formal meetings and through correspondence. The

[10, 13, 19] See list of research reports.

scope of the work included both the development of plans for laboratory work and the development of codes for testing and rating air cleaning devices for both ventilation and industrial practices. A code[7] developed by the committee for air cleaning devices used in general ventilation work was presented at the Semi-Annual Meeting 1933 and adopted with amendments for submission to the membership for approval by letter ballot.

Since the Detroit Meeting the committee has conducted investigations as to the desirability of adding a fibrous or linty constituent to the standard dust used in testing, and the possibility of more definitely defining particle sizes in the lamp black used in the standard dust. Professor Rowley, at the University of Minnesota in co-operation with the Laboratory, and S. R. Lewis, consulting engineer of Chicago, have studied the use of the test procedure for automatic air filters and have developed a performance chart. The University of Minnesota is continuing the study of the problems involved in cleaning the air of pollens and those dusts which cause allergic diseases such as hay fever and asthma.

In developing the code for air cleaning devices it became apparent that a code suitable for laboratory testing of air cleaning devices used in general ventilation work would be entirely unsuited for rating cyclone collectors, dust separators, and the like, or for the selection and rating of devices intended to supply air for dust-hazardous occupations in accordance with the standards of the U. S. Public Health Service. The use of the present code was, therefore, limited to laboratory investigation and rating of devices used in general ventilation work for the sole purpose of removing solid impurities from the air.

The committee has now under consideration a code for testing and rating air cleaning devices used in public health work. Prof. Philip Drinker has carried on preliminary work in the selection of testing dust having uniform particle sizes. The committee has received a number of requests for advice and assistance on procedures for testing air cleaners and has co-operated wherever possible.

4. CORRELATING THERMAL RESEARCH

R. M. Conner, *Chairman*

D. S. Boyden — H. T. Richardson
J. C. Fitts — Perry West

The work of cataloging available literature was continued at Pittsburgh and a new Directory[18] was published in conformity with earlier plans.

5. CORROSION

J. H. Walker, *Chairman*

H. F. Bain — W. H. Driscoll
E. L. Chappell — R. R. Seeber

The Technical Advisory Committee on Corrosion is advising Professor Seeber of the Michigan College of Mining and Technology regarding corrosion tests which are being undertaken under the direction of Dr. Rohrman of that institution, in co-operation with the Research Laboratory. The general plan of the work is to determine the rate of corrosion in various parts of a model heating system, operated under various conditions such as prevail in actual practice.

6. DIRECT AND INDIRECT RADIATION WITH GRAVITY AIR CIRCULATION

H. F. Hutzel, *Chairman* — J. P. Magos
A. P. Kratz — T. A. Novotney
H. R. Linn — R. N. Trane
J. F. McIntire — G. L. Tuve

As a result of increasing interest in the application of convector heaters, the following problems have been considered by this committee for possible investigation:

(1) Is the steam condensed in a convector a true measure of its heating ability in so far as comfort temperature is concerned?
(2) If not, what tangible means of measuring heating effect can be recommended?
(3) To what extent does height of enclosure affect heating effect?
(4) Does a top outlet enclosure produce a different heating effect than a side outlet enclosure?
(5) To what extent does the volume of air circulated affect the heating effect of a convector?
(6) Is the standard correction factor formula commonly used in the case of cast-iron direct radiation adaptable to convector types of heaters when tested in the warm wall testing booths as provided for in the code for testing convector heaters?

Of these phases of the problem two have been investigated during the past year at the University of Illinois as follows:

The investigation of direct radiators and convectors has proceeded along the lines previously adopted, with tests in both the room heating testing plant and the warm wall testing booth. In connection with the room heating testing plant the construction and calibration of the eupatheoscope was completed. A study of the application of this instrument led to the recommendation of a provisional standard for its use conforming with the standards of comfort accepted in American practice. A paper[8] on the subject was presented at the Semi-Annual Meeting of the Society.

The work in the warm wall testing booth led to the conclusions that the error in the application of the correction factor specified in the A. S. H. V. E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code) practically does not exceed 5 per cent when the temperature of the inlet air in the test is not more than 15 degrees above or 5 deg below the 65 F adopted as a standard. These results were presented[11] at the Semi-Annual Meeting 1933.

7. GAS HEATING EQUIPMENT

W. E. Stark, *Chairman* — Thomson King
Robert Harper — J. F. McIntire
E. A. Jones — H. L. Whitelaw

Consideration was given during the year to plans for a study of heat loss through draft hoods in gas heating installations. The work was outlined and data is being collected by the Chairman of the Technical Advisory Committee in Cleveland, and by gas companies in Chicago and Washington, D. C.

8. HEAT TRANSFER OF FINNED TUBES WITH FORCED AIR CIRCULATION

F. B. Rowley, *Chairman* — A. P. Kratz
H. F. Bain — E. J. Lindseth
H. F. Hutzel — G. L. Tuve
W. G. King — W. E. Stark

Plans for an investigation which is being carried on by Professors Tuve and McKeeman at Case School of Applied Science, Cleveland, in co-operation with the Society, were outlined by the committee and a test set-up was developed and data collected on the relation of heat emission to air circulation. A comparison of the findings with results obtained in similar studies made in England shows a greater variation with increased velocity in the case of the English studies and suggests that the minimum air velocity used in the study at Case School may not have been below the velocity of turbulence. This phase of the subject is now being investigated.

A paper for presentation to the Society is now in preparation at Case School which will present data answering the following questions:

1. If the performance of a finned-tube unit under one set of conditions is known, how can the performance under other conditions be predicted?
2. What effect has the character of the air stream upon the heat transfer?
3. Can the heat transfer of a finned-tube unit with forced air circulation be predicted by using the data already available in the publications of McAdams, Rowley, King, and others?

9. HEAT TRANSMISSION—(HEAT RECEIVED BY AND EMITTED BY BUILDINGS IN RELATION TO LIVING COMFORT)

P. D. Close, *Chairman*

A. B. Algren — A. E. Stacey, Jr.
R. E. Backstrom — J. H. Walker

Plans formerly outlined for investigations of this subject at Pittsburgh and at the University of Minnesota have been largely completed. A report[5] summarizing previous work at the University of Minnesota was published. Work now being carried on in heat transmission at Minnesota includes a more critical study of the thermal conductivity of special types of wall construction and an analysis of air space conductance. An attempt is being made to make a more definite separation between the total amount of heat conducted by radiation and by convection and conduction across the air space. The surface effect on both radiation and convection is being studied and some of the problems of practical application of insulation to air spaces are included.

The committee cooperated with the Technical Advisory Committee on Refrigeration in Relation to Air Treatment in a study carried on at Pittsburgh of solar radiation through windows and the effect of curtains, awnings and Venetian blinds in its reduction, which will be reported[20] at the Annual Meeting. A paper[14] relating to the amount of total low intensity radiant energy passing through various types of glass was prepared by

[7]See list of research papers.
[18]See list of research reports.

[5, 8, 11, 14, 20]See list of research reports.

the Research Laboratory and will also be given at the Annual Meeting, 1934.

10. INFILTRATION IN BUILDINGS

D. W. Nelson, *Chairman*
V. W. Hunter
W. C. Randall
E. N. Sanbern
J. G. Shodron
Ernest Szekely

There was a time when infiltration in buildings was regarded as a problem only during the heating season. With the widespread interest and adoption of summer cooling, the determination and control of infiltration becomes of year round importance. It is becoming increasingly necessary to know air leakage values and to improve the infiltration resistance of building units.

The proposed program of the Heat Transmission Committee points to the desirability of having a method of testing for infiltration through units such as windows in place in the rooms under investigation. This would be especially desirable in the testing of individual rooms on various floors of tall buildings. The variation in radiation required is likely largely due to a difference in infiltration. It is necessary that the windows in the various rooms selected have similar characteristics as to air leakage.

The fourteen double-hung wood windows used for the study on infiltration at the University of Wisconsin, in cooperation with the Laboratory and reported on in 1931, are installed in a shelter house. They are weatherstripped and have been removed once for testing. It is proposed to test them at intervals during a five-year exposure period to determine their effectiveness under actual building exposure and heating conditions.

11. OIL BURNING DEVICES

H. F. Tapp, *Chairman*
Elliott Harrington
F. B. Howell
J. H. McIlvaine
L. E. Seeley

Investigation of the plans outlined for studies of oil burners at Yale University, in cooperation with the American Oil Burner Association and the Society, was continued during the year at a reduced rate because of lack of sufficient funds. A large part of the work originally planned has been completed and results are contained in a report[15] being presented at the 1934 Annual Meeting of the Society. The program now being considered by the Committee includes the following studies:

1. Characteristics of a gravity hot water heating system in connection with oil burners.
2. Effect of furnace proportion on operation of various types of oil burners.
3. Effect of oil characteristics on oil burner performance.
4. Standard practice for determining fuel-burning rates of oil burners.
5. Application of burner sizes to heating load.
6. Boiler ratings for oil burner operation.

12. PIPE AND TUBING (SIZES) CARRYING LOW PRESSURE STEAM OR HOT WATER

S. R. Lewis, *Chairman*
J. C. Fitts
F. E. Giesecke
H. M. Hart
C. A. Hill
R. R. Seeber
W. K. Simpson

The laboratory work outlined was largely completed and a report[4] presented to the Society. Additional work remains to be done in the correlation of the results of the laboratory studies and their resulting application to the pipe size tables of THE GUIDE.

13. REFRIGERATION IN RELATION TO AIR TREATMENT

A. P. Kratz, *Chairman*
E. A. Brandt
E. D. Milener
K. W. Miller
F. G. Sedgwick
I. H. Walker
R. W. Waterfill

This project was outlined by the committee and the study carried on during the summer of 1932.[2, 3] The work was continued during the past summer with the object of determining to what extent ice cooling in the Research Residence could be supplemented by the circulation of cool air through the house from the outdoors during the hours from 6:00 p. m. to 6:00 a. m. and also the relative merits of several methods of circulating night air.

The results of the past summer's work indicate that if the best method of supplementary cooling with night air had been consistently used during the summer of 1932 the amount of ice required for that season would have been about 10 tons instead of the 43 tons used. Even with the least favorable method of night cooling the amount could have been reduced to 23 tons. The results will be presented[16] at the 1934 Annual Meeting of the Society. The National Warm Air Heating and Air Conditioning Association, The National Association of Ice Industries, The Utilities Research Commission of Chicago, and other organizations cooperated in this study.

The committee also cooperated with the Heat Transmission Committee in a study conducted at the Research Laboratory in Pittsburgh of solar radiation through windows and the effect of curtains, awnings and Venetian blinds in its reduction, a report[20] of which is being presented at the Annual Meeting, 1934.

14. SOUND IN RELATION TO HEATING AND VENTILATION

Warren Ewald, *Chairman*
C. A. Andree
Carl Ashley
C. A. Booth
V. O. Knudsen
R. F. Norris
J. P. Reis
G. T. Stanton

Consideration is being given to plans for research to standardize field methods of determining sound emanating from air conditioning equipment.

15. VENTILATION OF GARAGES AND BUS TERMINALS

E. K. Campbell, *Chairman*
S. H. Downs
T. M. Dugan
E. C. Evans
F. H. Hecht
H. L. Moore
A. H. Sluss

Plans previously outlined by the committee were investigated during the latter part of the last heating season and this fall, resulting in two publications.[9, 17]

16. HEAT REQUIREMENTS OF BUILDINGS

D. S. Boyden, *Chairman*
P. D. Close
W. H. Driscoll
H. M. Hart
P. E. Holcombe
V. W. Hunter
F. B. Rowley
R. J. J. Tennant
J. H. Walker

This committee was appointed during the year to coordinate and reconcile the many available data on heat transfer and loss, or gain, resulting from laboratory studies, with the actual formulae for using such values in the calculation of the heating requirements of a building. The work of the committee may be considered the final step in the Society's long and extended program of studying heat losses from buildings. Plans are being outlined for studies in Detroit, Pittsburgh, Minneapolis, Madison, Urbana, and other places, of the relation between calculated and actual heat requirements of buildings. At least some of these studies will include an accurate study of the hourly heat requirements to maintain desired temperature in individual rooms on different exposures and different elevations of modern buildings.

Projects now under way at the University of Wisconsin include a comparison of heating by means of direct radiation with heating by unit heaters, and studies of the economy of lowering temperatures during periods of unoccupancy and of the regulation of heat supply by zone control, by orifices, and by outside temperatures. These projects offer an excellent opportunity to obtain data useful to the Committee on Heat Requirements of Buildings concerning relations among calculated heat loss, installed radiation, actual heat loss, and the cooling-off and heating-up characteristics of building structures.

The economy of lowering temperatures during unoccupancy and the resulting peak loads on the vacuum pumps is being studied by the University of Wisconsin in cooperation with the Research Laboratory in the three-story, stone Mechanical Engineering Building which is heated by a vacuum system with direct radiation in offices and class rooms, unit heaters in most laboratories, and a central fan supply for class room ventilation. The system is equipped with a dual temperature control. During the present heating season a study will be made to determine the saving resulting from the carrying of reduced temperatures when not occupied. The maximum load occurs on the

[2, 3, 4, 15] See list of research reports.

[9, 16, 17, 20] See list of research reports.

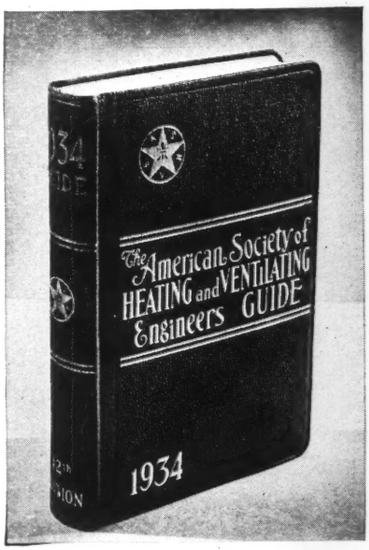
vacuum pump during the warming up periods and it is proposed to study these peak loads as related to the average load. The condensate and air removal loads will be studied separately. The discharge through calibrated orifices will be used to measure the air discharged.

The University of Wisconsin in cooperation with the Research Laboratory is making *a study of the regulation of heat supply by zone control, orifices, and by the outside temperature.* Tripp and Adams Halls are men's dormitories exactly similar except as to relation to points of the compass. A system of control of temperatures by zone valves and orifices and by radiator orifices has been installed in Adams Hall. An outside thermostat is used in determining the position of the zone valves. Condensate meters have been installed in both dormitories so that a comparison of steam consumptions may be obtained for a vacuum heating system and a similar system with the refinements of orifices and zone valves operating under control of outside temperatures.

Chadbourne Hall is a girls' dormitory heated by a one-pipe system. Condensate records have been kept over a period of years for this building. Recently a control of zone valves has been added that supplies steam intermittently in conformity to the heat losses as determined from the outside temperature. Provision is also made for reduced temperatures at night. Records are being made to determine the savings resulting from the added refinements in control.

The 12th Edition of the A. S. H. V. E. Guide

The A. S. H. V. E. Guide 1934 is just off the press and is ready for distribution. The Technical Data Section of this 12th Edition has been enlarged to include newly developed data that



The Guide 1934

are vitally important in meeting the present day demands of engineers who devote their time to heating, ventilating and air conditioning practice. From the practical experience of members as well as from available research sources useful facts have been gathered and incorporated in the 42 chapters which have been arranged for convenient reference. An extensive index has been included to aid the reader. The text is prepared for engineers, architects, contractors and students who are designing, operating, specifying, installing and studying systems and apparatus the functions of which are to create comfort and to improve the efficiency of processing.

All of the data in the previous edition have been reviewed, many chapters have been revised and amplified while others have been completely replaced. The new chapters include the Cooling Load and Cooling Methods, Unit Conditioners, Radiant and Electric Heating, Humidifying and Dehumidifying Equipment, Steam Heating Systems and Piping. Extensive changes will be noted in the chapters on Industrial Air Conditioning, Natural Ventilation, Central Fan Systems, Air Cleaning Equipment, Sound Control, Mechanical Furnace Systems, Radiators and Gravity Convectors, Heating Boilers, Pipe Insulation, Pipe, Fittings and Welding, Definitions and Terms. The remaining chapters were revised in order to bring them up-to-date.

Slight modifications were made in the chapters on Heat Transmission and Air Filtration based on a careful check of the experimental work which produced the basic figures. However the method of application in practice has been slightly modified in this 1934 edition of THE GUIDE.

With 42 chapters of 606 pages and a comprehensive index to the text, THE GUIDE 1934 presents the largest technical data section ever compiled and published by the Society. Supplemented by the engineering data of 108 leading manufacturers in the Catalog Section, which consists of 176 pages, the user will find essential facts that are invaluable in laying out a plant or selecting equipment.

THE GUIDE 1934 was prepared under the direction of THE GUIDE Publication Committee consisting of W. L. Fleisher, *Chairman,* D. S. Boyden, F. E. Giesecke, S. R. Lewis, Perry West and about fifty others cooperated in the preparation, reviewing and editing of the text.

Bound in a flexible red cover 6x9 in. in size, THE GUIDE 1934 contains 862 pages. Single copies are $5.00 each.

Western New York

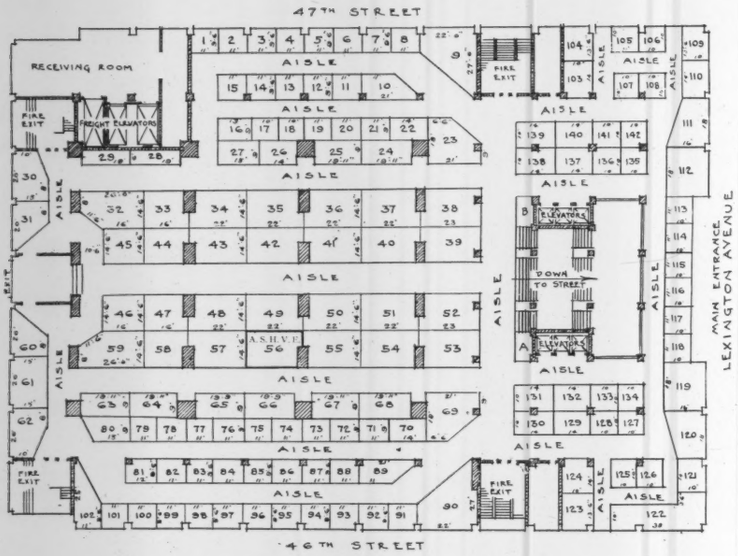
December 18, 1933. This meeting was the occasion of the Annual Xmas Stag Party and Pres. D. J. Mahoney, who had charge of the arrangements, made possible a very enjoyable evening at the University Club.

Approximately 35 members and guests vied for honors in bowling, bridge and other games, according to the report of Secy. W. E. Voisinet.

Roswell Farnham reported for the Nominating Committee, and, after a motion by Joseph Davis, the following slate was presented:

President—J. J. Yager.
1st Vice-President—W. F. Johnson.
2nd Vice-President—W. E. Voisinet.
Secretary—P. S. Hedley.
Treasurer—B. C. Candee.

At this meeting a number of the members signified their intention of attending the Society's Annual Meeting in New York, February 5-9.



Floor Plan, Grand Central Palace

125	Aerofin Corporation
97	Aerologist, The
49	Air Conditioning Industries, Inc.
10	Air Controls, Inc.
129	Allis-Chalmers Mfg. Co.
138	American Artisan
45	American Brass Co., The
9	American Gas Association, Inc.
36	American Gas Products Corp.
36–37–38–40–41	American Radiator Company
56	American Society Heating & Ventilating Engineers
5	Anderson Manufacturing Co.
72	Armstrong Machine Works
103	Automatic Products Co.
19–20	Barber-Colman Company
122	Baker Ice Machine Company, Inc.
67	Barnes & Jones, Inc.
78	Bell & Gossett Co.
101	Blom-Gamarra Humidifier
12	Breuer Electric Mfg. Co.
117	Bristol Company, The
9	Brooklyn Borough Gas Company
9	Brooklyn Union Gas Company
34–35	Bryant Heater Co., The
28–29	Burnham Boiler Corporation
30	Caloroil Burner Corporation
37	Campbell Metal Window Corp.
124	Carrier Engineering Corp.
76	Century Electric Co.
85–86	Century Engineering Corporation
75	Chace Valve Co., W. M.
55	Chase Brass & Copper Co., Inc.
10	Cleveland Heater Co.
126	Comfortaire Company
9	Consolidated Gas Company of N. Y.
81	Coppus Engineering Corp.
51	Crane Co.
38	Detroit Lubricator Co.
77	Dole Valve Co., The
115–116	Domestic Engineering Co.
21	Eagle-Picher Lead Co., The
89	Economy Pumping Machinery Co.
98	Emerson Electric Mfg. Co., The
40	Excelso Products Corp.
26	Fairbanks Company, The
107	Fairfield Oil Heating Co., Inc.
59	Fitzgibbons Boiler Co.
36 and 41	Fox Furnace Co.
133–134	Frick Company
74	Friez & Sons, Inc., Julien P.
32–33	Frigidaire Corporation
141–142	Fuel Oil Journal
69	General Electric Company (Air-Conditioning Div.)
111	General Electric Company (Oil Burner Div.)
23	General Gas Light Company
100	Hart & Cooley Manufacturing Co.
137	Heating Journals, Inc.
138	Heating, Piping & Air Conditioning
71	Heating & Ventilating
108	Heil Co., The

Exhibitors in Third International Heating and Ventilating Exposition

97	Hill Co., E. Vernon
126	Hinrichsen, Inc., A. F.
89	Hoffman Specialty Co., Inc.
132	Holtzer-Cabot Electric Co., The
87	Hotstream Heater Co., The
91	Independent Register & Mfg. Co.
3–4	Iron Fireman Mfg. Co.
16	Jenkins Bros.
62	Jones & Laughlin Steel Corp.
101	Kainer & Co.
88	Kelly, Inc., John G.
41	Kewanee Boiler Corp.
118	Knickerbocker Ice Company
6	Koven & Brothers, Inc., L. O.
136	Liquidometer Corp., The
139	McDonnell & Miller
114	Malleable Iron Fittings Co.
58	May Oil Burner Corp.
64	Mercoid Corporation, The
65–66	Minneapolis-Honeywell Regulator Co.
24	Modine Mfg. Co.
135	Monarch Manufacturing Works, Inc.
5	Moore Valve Corp., J. W.
121	Motometer Gauge & Equip. Corp.
61	Motorstokor Corporation
25	Mueller Furnace Co., L. J.
9	New York & Richmond Gas Co.
127–128	Niagara Blower Co.
11	Nu-Type Submerged Water Heater Corp.
137	Oil Heat
110	Orr & Sembower, Inc.
43	Owens-Illinois Glass Co.
130	Parker Appliance Co., The
48	Penn Electric Switch Co.
112	Petroleum Heat & Power Co.
47	Plumbing & Heating Trade Journal
57	Republic Steel Corp.
18	Reynolds Metals, Inc.
99	Rhoads & Sons, J. E.
52	Richardson & Boynton Co.
15	Richmond Radiator Co.
8	Russell & Co., Inc., W. A.
131	Sarco Co., Inc.
46	Scott-Newcomb, Inc.
30	Silent Glow Oil Burner Corp., The
73	Skidmore Corporation
50	Spencer Heater Company
57	Steel & Tubes, Inc.
80	Strang Air Conditioning Corp.
113	Taco Heaters, Inc.
123	Taylor Instrument Companies
63	Thatcher Co., The
17	Thrush & Co., H. A.
100	Tuttle & Bailey, Inc.
95–96	Uni-Flo Grille Corp.
44	United States Radiator Corp.
22	Vinco Co., Inc.
111-A	Volcano Burner Corporation
102	Waterloo Register Co.
88	Watts Regulator Co.
39	Warren Webster & Co.
42	Weil-McLain Company
90	Westinghouse Elec. & Mfg. Co.
84	Whitty Manufacturing Co., Inc.
93–94	Wing Mfg. Co., L. J.
79	Wickwire Spencer Steel Co.
140	Wizard Oil Burner Co.
54	Wood Hydraulic Hoist & Body Co.
53	York Ice Machinery Co.
82–83	York Oil Burner Co., Inc.
27	Young Radiator Company
68	Youngstown Sheet & Tube Co., The

Broadway at Night

Program

40th Annual Meeting

Feb. 5-9, 1934

Hotel Biltmore **New York**

9:00 a. m. Registration
10:00 a. m. Meeting of the Council
1:00 p. m. Opening of Third International Heating and Ventilating Exposition at Grand Central Palace

FIRST SESSION, Monday, February 5

2:00 p. m. Greeting
Response by Pres. W. T. Jones
Technical papers:
Automatic Controls for Forced Air Heating Systems—Konzo and Hubbard
Heating Buildings with Hot Water—Burt and Lewis
Report of the Council
Report of the Secretary
Amendments to By-Laws
Report of Finance Committee, John Howatt, Ch.
Report of Tellers of Election, E. E. Ashley, Ch.
7:00 p. m. Past President's Dinner (Transportation Club)
10:00 p. m. President's Reception and Get-Together Supper and Informal Dance, Paul Whiteman and his Orchestra, Casino Bleu, Hotel Biltmore
1:00 p. m. to 10:30 p. m. Heating and Ventilating Exposition

SECOND SESSION, Tuesday, February 6

9:30 a. m. Report of Committee on Research, Prof. G. L. Larson, Chairman
Technical Papers:
Design and Equipment of the Pierce Laboratory—C.-E. A. Winslow, Leonard Greenburg, L. P. Herrington and H. G. Ullman
Selecting Temperatures and Wind Velocities for Calculating Heating Losses, by P. D. Close
Radiation of Energy Through Glass—Blackshaw and Houghten
Studies of Solar Radiation Through Bare and Shaded Windows—Houghten, Gutberlet, and Blackshaw
Report of Committee on Code for Testing and Rating Condensation and Vacuum Pumps, John Howatt, Chairman
12:00 noon to 10:30 p. m. Heating and Ventilating Exposition

THIRD SESSION, Tuesday, February 6

2.00 p. m. Report of Guide Publication Committee, W. L. Fleisher, Chairman
Technical papers:
A Proving Home for Air Conditioning Investigations—Elliott Harrington and Leon A. Mears
Low-Cost Air Conditioning for a Small Residence —M. K. Drewry
Comfort Cooling with Attic Ventilating Fans—Helmrich and Tuttle
Study of Summer Cooling in the Research Residence for the Summer of 1933—Kratz and Konzo
2:30 p. m. Ladies' Bridge and Tea
7:30 p. m. Meeting Committee on Research
12:00 noon to 10:30 p. m. Heating and Ventilating Exposition.

FOURTH SESSION, Wednesday, February 7

9:30 a. m. Report of Membership Committee, E. K. Campbell, Chairman
Technical papers:
Thermodynamic Properties of Moist Air—Prof. J. A. Goff
Study of Air Conditioning and Office Employees Efficiency—McConnell and Kagey
Observations of Hay-Fever Sufferers in Air-Conditioned Room and the Relationship Between the Pollen Content of Outdoor Air and Weather Conditions—Kendall and Weidner
Corrosion Studies in Steam Heating Systems—Seeber, Rohrman and Smedberg
Report of Publication Committee, W. E. Stark, Chairman
12:00 noon to 10:30 p. m. Heating and Ventilating Exposition

FIFTH SESSION, Wednesday, February 7

2:00 p. m. Technical papers:
Carbon Monoxide Surveys of Two Garages—Sluss, Campbell and Farber
Diurnal and Seasonal Variations in the Small-Ion Content of Outdoor and Indoor Air—Yaglou and Benjamin
Air Conditioning in Its Relation to Human Welfare—Dr. C. A. Mills
Engineering in the Hospital of Tomorrow—Charles F. Neergaard
7:30 p. m. Annual Banquet and Dance, Meyer Davis Music, Ballroom, Hotel Biltmore

THURSDAY, February 8

9:30 a. m. Assemble at Radio City for Inspection of Mechanical Equipment in N. B. C. and British Buildings
11:00 a. m. Meeting of the Nominating Committee

SIXTH SESSION, Thursday, February 8

1:30 p. m. Ladies' Theater Party
2:00 p. m. Technical papers:
Study of Fuel Burning Rates and Power Requirements of Oil Burners in Relation to Excess Air —Seeley and Tavanlar
Design and Valuation of Cast Iron Domestic Heating Boilers—Dr. C. W. Brabbée
Conference on Nomenclature
Installation of Officers
Resolutions
Adjournment
4:00 p. m. Organization meeting of the Council
12:00 noon to 10:30 p. m. Heating and Ventilating Exposition, Grand Central Palace

FRIDAY, February 9

9:30 a. m. Meeting of Manufacturers of Heating, Ventilating and Air Conditioning Industry
Discussion of NRA Codes; Speakers, Beverly S. King, Asst. Deputy Administrator and Howard R. Heydon, Compliance Div., NRA.